NASA
SPACE VEHICLE
DESIGN CRITERIA
(CHEMICAL PROPULSION)

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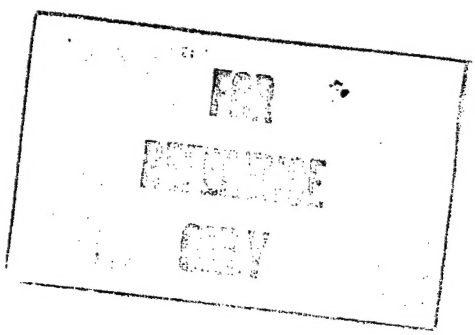
LIQUID ROCKET PRESSURE REGULATORS, RELIEF VALVES, CHECK VALVES, BURST DISKS, AND EXPLOSIVE VALVES



MARCH 1973

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technology:

Environment
Structures
Guidance and Control
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. This document, part of the series on Chemical Propulsion, is one such monograph. A list of all monographs issued prior to this one can be found on the final pages of this document.

These monographs are to be regarded as guides to design and not as NASA requirements, except as may be specified in formal project specifications. It is expected, however, that these documents, revised as experience may indicate to be desirable, eventually will provide uniform design practices for NASA space vehicles.

This monograph, "Liquid Rocket Pressure Regulators, Relief Valves, Check Valves, Burst Disks, and Explosive Valves", was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center, project management was by M. Murray Bailey. The monograph was written by L. E. Tomlinson of Rocketdyne Division, North American Rockwell Corporation, and was edited by Russell B. Keller, Jr. of Lewis. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, James V. Smith of Aerojet Liquid Rocket Company; Frank L. Merritt of TRW Systems, TRW, Inc.; and Eugene J. Fourney and Joseph M. Ladd of Lewis Research Center collectively and individually reviewed the monograph in detail.

Comments concerning the technical content of this monograph will be welcomed by the National Aeronautics and Space Administration, Lewis Research Center (Design Criteria Office), Cleveland, Ohio 44135.

March 1973

GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to organize and present, for effective use in design, the significant experience and knowledge accumulated in development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The monograph is organized into two major sections that are preceded by a brief introduction and complemented by a set of references.

The State of the Art, section 2, reviews and discusses the total design problem, and identifies which design elements are involved in successful design. It describes succinctly the current technology pertaining to these elements. When detailed information is required, the best available references are cited. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the *Design Criteria* and Recommended Practices.

The *Design Criteria,* shown in italics in section 3, state clearly and briefly what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The *Design Criteria* can serve effectively as a checklist of rules for the project manager to use in guiding a design or in assessing its adequacy.

The Recommended Practices, also in section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the *Design Criteria,* provide positive guidance to the practicing designer on how to achieve successful design.

Both sections have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

The design criteria monograph is not intended to be a design handbook, a set of specifications, or a design manual. It is a summary and a systematic ordering of the large and loosely organized body of existing successful design techniques and practices. Its value and its merit should be judged on how effectively it makes that material available to and useful to the designer.

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LIQUID ROCKET PRESSURE REGULATORS, RELIEF VALVES, CHECK VALVES, BURST DISKS, AND EXPLOSIVE VALVES

1. INTRODUCTION

Regulators, relief valves, check valves, burst disks, and explosive valves are used extensively in liquid propellant rocket systems. These fluid-control devices are found in pressurizing systems for propellant feed of reaction controls, in main-engine control systems, and in vehicle tank pressurization systems. In the Saturn V vehicle, for example, there are 34 regulators, 62 relief valves, 286 check valves, and 64 burst disks, including explosive units. Regulators reduce the pressure and control the flow of the gas to meet the demands of the system. Relief valves protect the elements of both high- and low-pressure systems from inadvertent overpressurization and failure. Check valves are unidirectional flow devices used in control functions. Burst disks are passive flow-blocking members in a fluid-carrying line; they initiate flow when ruptured by line pressure. Explosive valves are valving devices that are actuated by a small explosive charge; they can be either normally open or normally closed before actuation.

Many problems in the design and operation of these devices in space vehicles are created by the stringent operational requirements and the severe service environments. High vibrational loads have caused structural failures, loosened joints, and shifted operation beyond acceptable limits. Low operational temperatures require that components be protected from freezing moisture; special seals are required, and usable materials are limited. Other operational problems have included unstable operation, slow response, regulated-pressure overshoot and undershoot, fluid contamination, component/fluid compatibility, leakage, failure to operate, and poor repeatability. The current proven reliability of these components in liquid rocket systems demonstrates the benefits of careful investigation to identify failure modes and extensive testing to prove the adequacy of the problem solution.

This monograph is based on a critical evaluation of the experiences and practices in the design, test, and use of these control devices in operational space vehicles. The material in the monograph is organized for natural and effective use by designers. Each of the five devices is treated separately. The discussion on regulators includes selection of type of assembly; detail problems with inlet valves, actuators, and control elements; and operational problems. Relief valves contain many items similar to regulators, and therefore the

discussion covers the same topics; in addition, because tank relief valves frequently are combined with a tank venting function, some of the venting problems are included. The check valve discussion covers the two basic types, poppet and flapper, and describes problems resulting from the use of the check valve in space vehicle systems. The various problems encountered in the design and application of burst disks are discussed. Explosive valves are specialized units with a limited source of manufacturers who also make the design; most of the discussion of this component therefore is concerned with operational problems. For all of the units, the choice of optimum configuration is important, and guides to aid in configuration selection are outlined.

2. STATE OF THE ART

The technology for aerospace regulators, relief valves, and check valves originated in commercial industry, where these units are used extensively; burst disks and explosive valves have the same origin but are used on a limited scale. For aircraft applications, the rugged, heavy commercial units were modified to reduce size and weight without loss of reliability and long life. The early space vehicles and engines employed many of the aircraft hydraulic-type controls modified for rapid response and operation in extreme vibration environments. The requirements of cryogenic[1] oxidizers led to the development of new sealing methods and the use of high-pressure stored helium or nitrogen for pressurization and controls functions. In the upper stages of the Saturn vehicle, for example, the controls and pressurization systems except for the hydraulic-controlled gimbaling system utilize helium. The advent of storable propellants necessitated the development of control components using materials compatible with these exotic fluids.

Regulators for space applications include single-stage, piloted, and series arranged units that may have shutoff or lockup capabilities. Flow capacities have varied from very low flows such as 0.002 lb/sec (0.91×10^{-3} kg/sec) to very high flows such as 10.0 lb/sec (4.54 kg/sec) with many types of gaseous media. Relief valves include direct-operated, inverted-operation, and pilot-operated units; they may have redundant pilots, overrides to open and close, and multiple cracking pressures. Flow capacities vary from 0.0065 lb/sec (2.95×10^{-3} kg/sec) of helium to 50.0 lb/sec (22.7 kg/sec) of gaseous oxygen. Check valves vary in size and type and may be dual or quad arrangements for redundancy in critical application. Burst disks in general are used for isolation or for zero-leakage applications. Explosive valves are used in locations where fast, positive valve action is required. Tables I through IV list the characteristics of some of the regulators, relief valves, check valves, and burst disks used in space applications.

The basic design of space-type regulators and relief valves is essentially the same as that of industrial types. The main differences are in the packaging and special features required to maintain narrow-band regulation when input is changing rapidly and the unit is exposed to environmental extremes. The interaction of components in complicated pneumatic control systems is studied with simulators using actual components and with computerized mathematical models. These tools are very useful in failure analysis and in solving stability problems. To optimize the units, tradeoff studies are performed to select the configuration that is best adapted to the system requirements and functional operation. The units in most space applications are highly specialized and extensively tested to ensure reliable operation. Components used in space applications are described in references 1 through 4; those for commercial applications are treated in references 5 through 14.

[1]Terms, symbols, and materials are defined in the Glossary.

Table I. – Design Features of Representative Operational Pressure Regulators

Vehicle or System	Application	Fluid Controlled	Supply pressure (max./min.) psi	Supply pressure (max./min.) MN/m^2	Regulated pressure psi	Regulated pressure MN/m^2	Flowrate lbm/sec	Flowrate kg/sec	Operating Temperature F°	Operating Temperature K
Jupiter	Guidance	N_2	3000/200	20.68/1.38	28 to 45	0.193 to 0.310	0.024	0.0109	-65 to +125	219 to 325
	Roll control	N_2	3000/500	20.68/3.45	300±5.0	2.07±0.034	0.3	0.14	-65 to +125	219 to 325
Thor	Engine control	N_2	3000/1000	20.68/6.895	640±25	4.41±0.17	1.0	0.454	-65 to +160	219 to 344
	Fuel tank pressurization	He	3500/700	24.13/4.83	25±1.5	0.17±0.010	0.026	0.0118	-100 to +160	200 to 344
Atlas	Engine control	He	3000/1000	20.68/6.895	790±25	5.44±0.17	0.3	0.14	-65 to +160	219 to 344
	Fuel tank pressurization	He	3100/110	21.37/0.758	64 to 67	0.44 to 0.46	1.0	0.454	-65 to +450	219 to 506
	Oxidizer tank pressurization	He	3100/75	21.37/0.517	23 to 36	0.16 to 0.25	1.0	0.454	-65 to +450	219 to 506
Titan I (1st stage)	Fuel tank pressurization	He	3250/200	22.41/1.38	12.0±0.5	0.083±0.003	0.1	0.0454	-260 to +290	111 to 417
	Oxidizer tank pressurization	He	3250/200	22.41/1.38	34.0±0.5	0.234±0.003	0.11	0.0499	-260 to +290	111 to 417
Agena	Attitude control: High mode	N_2 or CF_4	3600/300	24.82/2.07	105±5	0.724±.035	0.44	0.200	-90 to +200	206 to 367
	Low mode	N_2 or CF_4	3200/50	22.06/0.345	4.5 to 6.0	0.031 to 0.041	0.018	0.00817	-90 to +200	206 to 367
Delta	Attitude control	N_2	4000/375	27.58/2.59	300±30	2.07±0.21	0.013 to 0.38	0.00590 to 0.172	-50 to +160	228 to 344
	Tank pressurization	He	4200/400	28.96/2.76	345±10	2.38±0.069	0.084	0.0381	-65 to +175	219 to 353
Centaur	Engine control	He	3360/700	23.17/4.83	440 to 475	3.03 to 3.27	0.0025	0.00113	-5 to +125	253 to 325
	H_2O_2 subsystem	He	440/375	3.03/2.59	297 to 315	2.05 to 2.17	0.0025	0.00113	-5 to +125	253 to 325
Saturn S-1C	Control pressure and oxidizer purge	He	3000/1000	20.68/6.895	750±25	5.17±0.17	0.3	0.14	-65 to +160	219 to 344
S-II	Engine control	He	3000/600	20.68/4.14	400±25	2.76±0.17	0.1	0.0454	-200 to +140	144 to 333
	Fuel tank pressurization	H_2	860	5.93	28.5 to 30.0	0.196 to 0.207	2.4 to 4.5	1.09 to 2.04	-290 to +160	94 to 344
	Control pressure, fuel and oxidizer	He	3050/850	21.03/5.86	675 to 800	4.654 to 5.516	0.00167	0.0007575	–	–
	Oxidizer tank pressurization	O_2	1100	7.584	36.0 to 37.5	0.248 to 0.259	4.5 to 10.5	2.04 to 4.763	-180 to +860	156 to 733
	Helium injection, fuel	He	3050/1000	21.03/6.895	625 to 750	4.31 to 5.17	0.0083	0.00377	–	–
S-IVB	Fuel tank vent regulator	He	3100/400	21.37/2.76	250±2	1.72±0.014	0.18	0.0817	-423 to +160	21 to 344
	Control pressure, fuel	He	3100/600	21.37/4.14	475±25	3.28±0.17			-125 to +165	186 to 347

	Cold helium, oxidizer	He	3000/500	20.68/3.45	385±25	2.65±0.17	0.45	0.204	-420 to +160	22 to 344
	Auxiliary power system control: Primary	He	3200/350	22.06/2.41	196±3	1.35±0.021	0.0033	0.00150	-65 to +110	219 to 317
	Secondary	He	3200/350	22.06/2.41	200±3	1.38±0.021	0.0033	0.00150	-65 to +110	219 to 317
Lunar Modual	Fuel and oxidizer tank pressurization	He	1700/320	11.72/2.21	240 to 248	1.655 to 1.710	0.00867 to 0.0917	0.00393 to 0.0416		
Service Module	Fuel tank pressurization: Primary	He	3500/400	24.13/2.76	180 to 188	1.241 to 1.296	0.0242	0.0110	-65 to +160	219 to 344
	Secondary	He	3500/400	24.13/2.76	186 to 194	1.282 to 1.338	0.0242	0.0110	-65 to +160	219 to 344
	Oxidizer tank pressurization: Primary	He	3500/400	24.13/2.76	170 to 180	1.172 to 1.241	0.0242	0.0110	-65 to +160	219 to 344
	Secondary	He	3500/400	24.13/2.76	178 to 186	1.227 to 1.282	0.0242	0.0110	-65 to +160	219 to 344
Command Module	Reaction controls: Primary	He	4500/400	31.03/2.76	291±4	2.006±0.028	0.005	0.0023	-65 to +110	219 to 317
	Secondary	He	4500/400	31.03/2.76	295±4	2.034±0.028	0.005	0.0023	-65 to +110	219 to 317
Nimbus	Attitude control	N_2	3900/200	26.89/1.38	35±2	0.241±0.014	0.0063	0.00286	-60 to +200	222 to 367
		Freon #4	4800/500	33.10/3.45	380±20	2.620±0.138	5.1	2.31	-100 to +170	200 to 350
Mercury Gemini	Life support	O_2	7500/200	51.71/1.38	80±10	0.552±0.069	0.004	0.0018	-80 to +200	211 to 367
Surveyor	Fuel and oxidizer pressurization	He	5200/820	35.85/5.65	725±20	5.00±0.14	0.009	0.0041	-150 to +115	172 to 319
	Fuel cell	H_2	30,000/25	206.85/0.172	1.0±0.2	0.0069±0.0014	6.13×10^{-7}	2.78×10^{-7}	-100 to +160	200 to 344
NASA Reentry	Electronics pressurization	He	5000/200	34.48/1.38	1.5±0.075	0.010±0.0005	0.00061	0.00028	-80 to +160	211 to 344
Ranger	Cold gas attitude control	N_2	4000/150	27.58/1.03	15±1	0.10±0.0069	0.0015	0.00068	-10 to +160	250 to 344
Mariner	Cold gas attitude control	N_2	4000/150	27.58/1.03	15±1	0.10±0.0069	0.0015	0.00068	-10 to +160	250 to 344
	Fuel tank pressurization	N_2	3600/400	24.82/2.76	308±3	2.12±0.021	0.006	0.0027	+14 to +167	263 to 344
Lunar Orbiter	Cold gas attitude control	N_2	4000/150	27.58/1.03	19.5±10	0.134±0.069	0.0027	0.0012	-15 to +160	247 to 344
	Propellant tank pressurization	N_2	3820/650	26.34/4.48	180 to 200	1.24 to 1.38	0.0071	0.0032	-65 to +125	219 to 325
Pioneer	Cold gas attitude control	N_2	4000/200	27.58/1.38	50±2.5	0.34±0.017	0.0022	0.0010	-10 to +160	250 to 344

Table II. – Design Features of Representative Operational Relief Valves

Vehicle or System	Application	Fluid Controlled	Pressure (full flow/reseat) psi	Pressure (full flow/reseat) MN/m^2	Full flow lbm/sec	Full flow kg/sec	Operating Temperature °F	Operating Temperature K
Thor	Engine control	N_2	800/600	5.52/4.14	1.0	0.454	-65 to +160	219 to 344
	Fuel tank	RP-1	38/32	0.26/0.22	1.33	0.603	-65 to +160	219 to 344
	Oxidizer tank	O_2	38/33	0.26/0.23	20.0	9.07	-310 to +160	83 to 344
Atlas	Engine control	N_2	900/750	6.21/5.17	0.3	0.14	-65 to +160	219 to 344
	Fuel tank	He	71.7/68.5	0.494/0.472	0.5	0.23	-100 to +450	200 to 506
	Oxidizer tank	He	39.3/36.6	0.271/0.252	0.5	0.23	-100 to +450	200 to 506
Agena	Attitude control	N_2	225/150	1.55/1.03	0.45	0.204	-90 to +200	205 to 367
		CF_4	225/150	1.55/1.03	0.128	0.0581	-90 to +200	205 to 367
Centaur	Engine control	He	525/475	3.62/3.28	0.1	0.0454	-100 to +200	200 to 367
	H_2O_2 subsystem	He	354/320	2.44/2.21	0.1	0.0454	-100 to +200	200 to 367
Saturn S-IC	Fuel tank	He	35/30.5	0.24/0.21	3.0	1.36	-150 to +165	172 to 347
	Oxidizer tank	O_2	25.5/24.0	0.176/0.165	50.0	22.7	-320 to +450	78 to 506
S-II	Engine conrol	He	4000/3450	27.58/23.79	0.0065	0.0029	-200 to +160	144 to 344
	Fuel tank	He	550/530	3.79/3.65	0.1	0.0454	-200 to +160	144 to 344
	Fuel controls	H_2	29.5/27.5	0.203/0.190	5.5	2.49	-423 to +160	20 to 344
	Oxidizer controls	He	825/760	5.69/5.24	0.1	0.0454	-180 to +160	155 to 344
	Oxidizer tank	O_2	42.0/39.0	0.290/0.269	18.0	8.16	-297 to +160	90 to 344
S-IVB	Fuel tank	H_2	34.0/31.0	0.23/0.21	1.5	0.680	-420 to +160	22 to 344
	Cold helium relief	He	3500/3100	24.13/21.38	0.67	0.304	-420 to +160	22 to 344
	Auxiliary power system control	He	375/325	2.59/2.24	0.104	0.0472	-40 to +160	233 to 344
Lunar Module	Fuel and oxidizer tank pressurization	He	275/254	1.90/1.75	0.167	0.0758	–	–
Service Module	Fuel and oxidizer tank pressurization	He	250/225	1.72/1.55	0.067	0.0304	–	–

Table III. – Design Features of Representaive Operational Check Valves

Vehicle or System	Application	Fluid checked	Fluid flow-through	Cracking Pressure psid	Cracking Pressure MN/m^2	Loading method	Type
Atlas	Engine control	Lox	He	10	0.069	Spring	Poppet
		Lox	Lox	2	0.014	Spring	Flapper
Thor	Engine control	RP-1	N_2	10	0.069	Spring	Poppet
Saturn	Engine control	Lox	He	10	0.069	Spring	Floating soft poppet
		H_2	H_2	15	0.103	Spring	Floating soft poppet
		RP-1	Glycol	20	0.183	Spring	Poppet
		Lox	Lox	1	0.0069	Spring	Flapper

Table IV. – Design Features of Representative Operational Burst Disks

Vehicle or System	Application	Fluid controlled	Burst Pressure psi	Burst Pressure MN/m^2	Material	Disk Type	Installation
Atlas	Hypergol igniter	TEAB, RP-1	300^{+25}_{-70}	$2.07^{+0.172}_{-0.483}$	6061-0	Coined	Welded
	Solid propellant gas generator	Hot gas	650±50	4.48±0.345	3003-0	Coined	Mechanically retained
Titan	Autogenous pressurization	N_2O_4 vapor	300±30	2.07±0.207	1100-0	Coined	Mechanically retained
		A-50 vapor	300±30	2.07±0.207	1100-0	Coined	Mechanically retained
Saturn	Hypergol igniter	TEAB, RP-1	500^{+25}_{-75}	$3.45^{+0.172}_{-0.517}$	6061-0	Coined	Welded
		TEAB, RP-1	300^{+25}_{-75}	$2.07^{+0.172}_{-0.517}$	6061-0	Coined	Welded
	Pump drain line	O_2	21±0.5	0.145±0.0034	1100-0	Coined	Mechanically retained

2.1 REGULATORS

The regulators in space applications are almost always special units designed specifically to meet unique requirements. The rigid requirements, with particular emphasis on size, weight, and efficient operation of the total system, have limited the use of readily available commercial units. For example, when a commercial unit was used on an engine for the Atlas vehicle, special procurement controls were necessary because product improvements or material changes acceptable for most industrial usages could not be accepted for use in the environment of the propulsion system. Once selected, a design configuration for space use must not change in any way that is not planned or tested.

In large-engine control systems, the regulator maintains a constant-pressure source of gas to operate engine control valves, main propellant valves, gas-generator valves, start systems, and other control components. In systems using a pressurized, noncryogenic propellant for hydraulic controls, the components are designed to operate at the variable pump output pressure, and a pressure regulator is not required. In the pneumatic control systems, variations of regulated pressure will affect valve timing and sequencing in engine start and shutdown operation. On a Saturn engine, for example, a valve leaking high-pressure gas into the helium control system resulted in an abnormally high control pressure that so altered the shutdown timing that the engine was damaged seriously.

In reaction-control systems, the energy source for propellant feed is supplied by high-pressure stored gas. The regulator maintains the desired pressurizing flow and pressure to the propellant tanks as the stored-gas pressure decreases. The regulated-pressure tolerance band is narrow and, in order to utilize as much of the stored gas as possible, the minimum supply pressure approaches regulated pressure. If the regulator is required to lock up, i.e., shut off at a pressure above regulated pressure, the lockup pressure is established a minimum above the maximum regulated pressure, and allowable leakage is kept small to limit gas loss during vehicle coast time.

In large space vehicles, the propellant tanks are pressurized to provide a positive pump suction head. For noncryogenic propellants, the high pressure of bottled gas is reduced by a regulator to the level desired for pressurizing the tanks. Cryogenic propellant tanks are pressurized by the propellant boiloff and, where necessary, by the addition of warm gaseous propellant. This gaseous propellant is obtained by flowing high-pressure liquid propellant through a heat exchanger, the pressure and flowrate being controlled with a regulator.

2.1.1 Assembly

Modulating pressure-reducing regulators are designed to maintain a constant pressure at a chosen set point; they may be categorized as direct-acting, dome-loaded, or pilot-operated. In the direct-acting regulator, the inlet valve is positioned by a sensor (or actuator) with a reference load (usually a mechanical spring) for motion in one direction and a

regulated-pressure force acting on the sensor for motion in the opposite direction. In the dome-loaded series-arranged regulator, the reference load to position the actuator and inlet valve is a constant-pressure reference that may be supplied from an external source or from the regulator supply-pressure source reduced to the desired constant reference pressure. In the piloted regulator, the force to position the actuator and inlet valve is a proportional pressure force controlled by a pilot valve that is positioned by a reference load and sensed variations in regulated pressure.

Nonmodulating regulation systems are designed to maintain pressures between chosen limits; a signal from a pressure switch that senses variations in tank pressure, for example, opens and closes valves that control the input of the pressurizing gas. The tank pressure varies between the pressure-switch settings and the additional pressure change resulting from rate of change of tank pressure and the response time of the pressure switch and valve. Thus, the tank pressure continually cycles between the maximum and minimum pressure levels at rates proportional to the gas flowrate and propellant flowrate.

As a first step in design, the designer selects the type of regulator best suited for the application. All the design requirements, operational characteristics, and environmental restraints must be established before the valve configuration is selected. The orientation, acceleration loads, shock inputs, and vibrational levels of the vehicle in which the component is used may be critical to the operation of the unit. Costly redesigns have resulted from incompletely defined requirements and a poor choice of configuration. When there is insufficient time to redesign, a unit may be modified but still be marginal for the application.

The choice of a modulating regulator assembly is a progressive procedure. Although the experienced designer intuitively may check only a limited approach to obtain a solution, many considerations are involved in selecting the appropriate type of regulator. The designer normally starts with the simplest type that could be used; if it fails to meet the design requirements or proves to be excessively large for the system, he proceeds to the next level of complexity. The effect of different factors on modulating regulator complexity is given in figure 1. The direct-acting regulator is the simplest, with the fewest moving parts and leakage paths. The piloted regulator generally is the most complicated, with many moving parts and leakage paths. The simplest configuration will tend to have greatest reliability and lowest cost and weight. The system in which the regulator is to be used is reviewed periodically to determine whether the desired regulator performance can be relaxed. It may be possible to use a less complicated design by changing the specified requirements in a manner that does not affect the system seriously. The final design represents the best available combination of performance, size, weight, and cost.

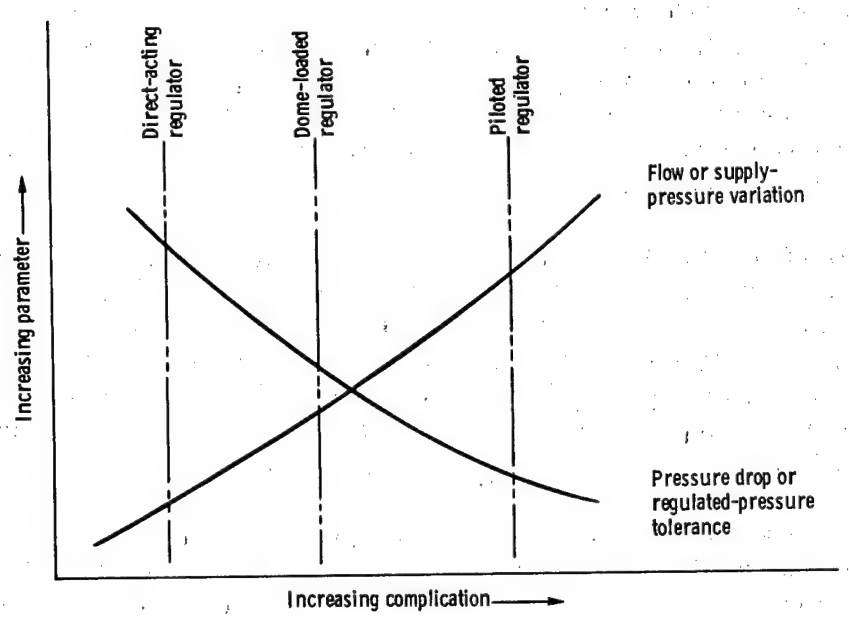


Figure 1. - Factors in regulator complexity.

2.1.1.1 DIRECT ACTING

A typical direct-acting regulator is shown in figure 2. After the inlet valve size, stroke, and type have been established (sec. 2.1.2), the actuator (or sensor) size is determined. The principal factor in sizing the actuator is the requirement to limit the regulated-pressure error due to supply-pressure variation and flow demand (refs. 15 through 18). For steady-state operating conditions, the regulated-pressure error per square inch of actuator area is determined for the required supply-pressure variation and flow demand. The actuator and its reference spring are sized to limit these errors within the allowable regulated-pressure tolerance. The resulting actuator and spring may be excessively large and the assembly may be prone to malfunction when operating in vibration environments. When a given actuator and reference spring become too large, a different type of regulator may be chosen in order to reduce weight and improve operation.

As a general practice, the regulated-pressure error due to supply-pressure variation and flow demand is limited to approximately 25 percent of the total allowable regulated-pressure variation. Other sources of regulated-pressure errors include variation in temperature, sustained acceleration, vibration, mechanical hysteresis, friction, and manufacturing tolerances including spring-rate variation.

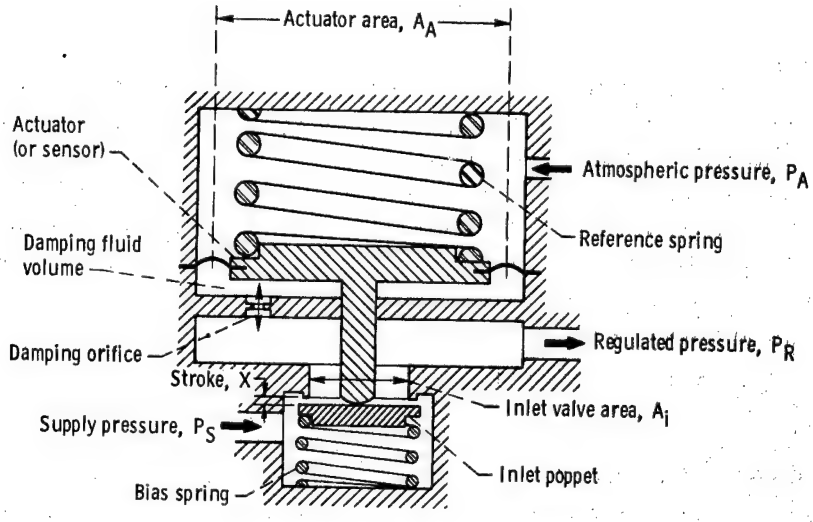


Figure 2. - Schematic of typical direct-acting regulator.

2.1.1.2 DOME LOADED

The dome-loaded configuration (fig. 3) uses a constant-pressure reference source and actuator rather than the large reference spring to supply an actuation force for the inlet valve. As the required flow demand increases, the corresponding inlet valve size increases, and a large actuator is needed to limit the regulated-pressure error resulting from supply-pressure variation and flow demand. An external reference source of constant-pressure gas is not readily available in space applications, and a small-capacity direct-acting pressure-unbalanced regulator (loader valve) is used to reduce the supply

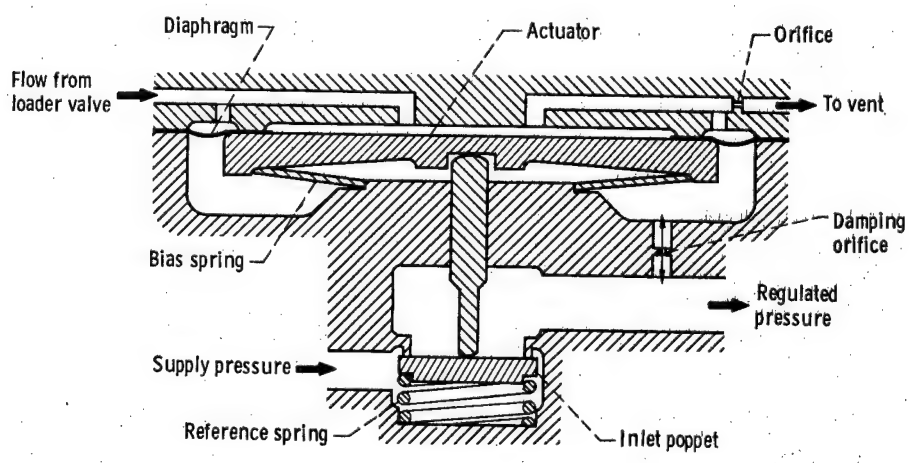


Figure 3. - Schematic of typical dome-loaded regulator.

pressure to the desired reference pressure: With this arrangement, the errors of the small loader valve are established such that reasonable sensor size and reference-load spring may be used, and the main inlet valve actuator is sized to keep the total error of both regulators within prescribed limits. This arrangement is adaptable to systems in which the small loader valve either exhausts externally or locks up. Once again, if the springs, sensor, and actuator necessary to keep the regulated pressure within specified limits become large, a different type of regulator must be considered.

2.1.1.3 PILOTED

In the piloted configuration (fig. 4), a relatively constant flow of control-pressure gas (bleed flow) is supplied to the inlet valve actuator. The pilot valve or controller senses the regulated

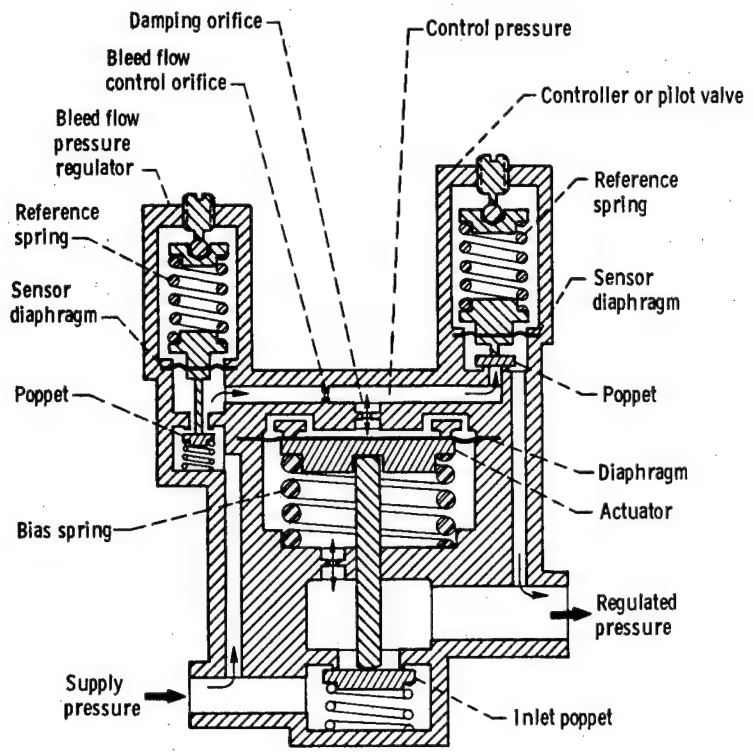


Figure 4. - Schematic of typical pilot-operated regulator.

pressure and varies the bleed flow out of the actuator, the resulting change in control pressure being proportional to changes in regulated pressure. The inlet valve is positioned to meet the regulator flow demand. The regulated-pressure error is a function of the gain in the pilot circuit with its variables and the inlet valve errors. The control pressure may be

supplied by a bleed regulator that reduces the supply pressure to a relatively constant pressure. The bleed-flow control orifice downstream of the bleed regulator controls the quantity of bleed flow. Acceptable operation of the pilot-operated regulator depends on the correct sizing of the elements.

The configuration is analyzed at steady-state operating conditions to establish the approximate sizes, strokes, and mechanical characteristics of the elements (refs. 19 and 20). This steady-state analysis is followed by a dynamic analysis to verify that the response characteristics are satisfactory and the operation is stable (refs. 21 and 22).

2.1.1.4 NONMODULATING REGULATION SYSTEM

A nonmodulating pressure-regulating system, as used for pressurizing large tanks, consists basically of a pressure switch for sensing tank pressure and an on-off valve for controlling the pressurant flow. (A pressure transducer with an amplifying circuit may be substituted for the pressure switch). When the tank pressure decreases, the electrical contacts of the pressure switch close, and an electrical signal opens an electrically operated shutoff valve; pressurant flows to the tank to increase the tank pressure. As the desired maximum tank pressure is approached, the pressure-switch contacts open, the shutoff valve is deenergized, and the pressurant flow stops. Thus the tank pressure varies between a maximum and minimum with a cyclic period dependent upon the rate of change of tank pressure due to tank fluid flowrate, pressurant flow, and operating time of the pressure switch and shutoff valve. An analytical solution for the expected regulated-pressure variation of this type of system is given in reference 23; the results of a series of tests of a nonmodulating system are presented in reference 24. This system can be modified by providing several shutoff valves that are connected in parallel and progressively opened to supply an increasing flow of pressurizing gas as the tank ullage increases. The Centaur upper stage, for example, utilizes a quadruple redundant switch arrangement that has been very effective.

A nonmodulating system requires an external source of energy. Failure of the power source or any of its components can cause a failure of the pressurizing system. Regulated pressure with a narrow tolerance band is difficult to achieve because it is dependent upon the pressure-switch differential between open and close pressures. Some space vehicles have employed this type of regulation for tank pressurization, but most vehicles have used modulating regulators.

2.1.2 Inlet Valve

2.1.2.1 SIZE AND STROKE

The inlet valve of a regulator is positioned to control the fluid flow demand of a system and to maintain the required regulated pressure of that system. It is essential that the flow area

be adequate but not oversize; the inlet valve actuator and control elements are proportionally larger for an oversize inlet valve and can increase the size and weight of the unit. The flow area required for the rated flow at the most adverse operating conditions is determined as an equivalent orifice size from the equation for isentropic compressible flow (ref. 25). The inlet valve seat diameter and stroke are sized to provide this necessary flow area. Each type of valve – ball, cone, or flat poppet – requires a different maximum stroke and seat diameter. Stroke limitations of diaphragm-type actuators may determine the maximum possible stroke and influence the type of valve selected.

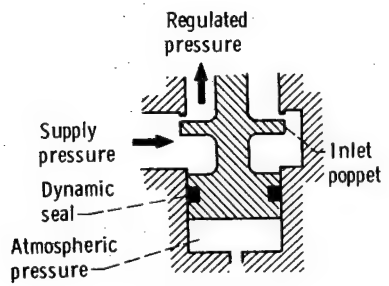
Regulator inlet valve design is determined by flow demand, pressure drop, leakage requirements, type of fluid, temperature extremes, and the preference of the designer. Successful designs have used a ball poppet on a sharp corner seat, a cone-shaped poppet on a cone-shaped seat, flat poppet and seat, trapped O-ring on a sharp corner seat, and a rotating disk and seat (butterfly). The poppets are guided over their travel by pistons sliding in bores; when balls are used, special piston-type followers retain the balls and slide in guiding bores. Clearance of the piston and bore, the normality of flat poppets or concentricity of conical poppets, and the indexing of mating parts must be checked to ensure that the poppet contacts the seat evenly and in alignment. The friction of these pistons dampens motion and introduces hysteresis to valve operation. Butterfly valves require seals on the rotating shaft; these seals can introduce design problems. Reference 26 presents details on valve poppet and seat characteristics and general valve design.

2.1.2.2 FLOW PASSAGES

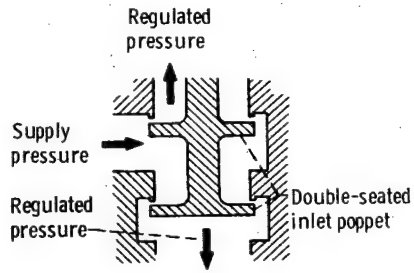
The type of inlet valve and its size and stroke are determined as described above. The flow passages from the inlet to the outlet of the regulator are critical; the pressure drop in these passages is held as small as possible so that a maximum amount of the allowable pressure drop is available for regulation by the inlet valve. The effects of the pressure drop of an inlet filter are considered. Pressure-flow characteristics of pneumatic valves are given in reference 27. If the passages downstream of the inlet valve are restricted and the flow of gas at the downstream point is sonic, the inlet valve will not regulate the pressure as required; the effects of restricted regulator outlets are described in references 28 and 29.

2.1.2.3 PRESSURE BALANCING

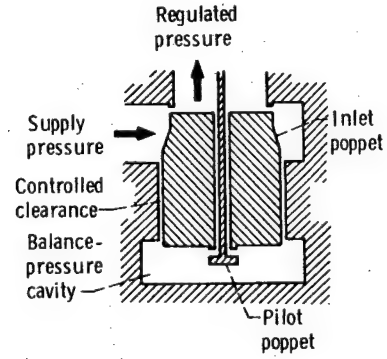
Pressure-balanced inlet valves are employed to decrease regulated-pressure errors and actuator size; the best use is in applications requiring large flows with very accurate pressure regulation (35.0±0.5 psig [241.3±3.4 kN/m^2]). Four methods of balancing poppet-type inlet valves are shown in figure 5. Although the dynamic seal (fig. 5(a)) is the simplest arrangement, the inherent hysteresis and friction of the sliding seal limits the use of this method of pressure balancing (ref. 30). The double-seated inlet valve (fig. 5(b)) is difficult to design and expensive to manufacture because the two poppets must contact the seats simultaneously for sealing. Experience has demonstrated that the piloted closed-loop system (fig. 5(c)) is basically unstable, i.e., the regulated pressure oscillates because of the high gain



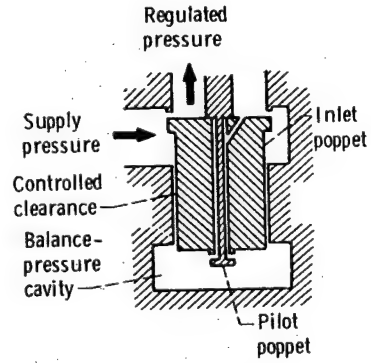
(a) Dynamic seal.



(b) Double seated.



(c) Piloted closed loop.



(d) Piloted open loop.

Figure 5. - Methods of balancing inlet valves.

of the system. For stable operation, Coulomb-damping rings are used on the inlet poppet to reduce hunting of the poppet; the frictional drag of these rings reduces the accuracy of regulation. The hunting also may be reduced by providing a large balance-pressure cavity. Because this cavity is exposed to the high supply pressure, the size and weight of the regulator is increased. In the piloted open-loop arrangement (fig. 5(d)), the initial pilot motion bleeds down the balance pressure. Additional force must be provided to open the inlet poppet, because the balance pressure exceeds the regulated pressure and the effective areas upon which they act are essentially equal. The open-loop arrangement is stable, and small flow demands are satisfied by the pilot valve stroke. When closed, both of the piloted inlet valves are pressure-unbalanced for sealing. The design involves numerous details and manufacturing difficulties including the controlled clearance of the inlet poppet in its guiding bore; consequently, there is limited use of piloted inlet valves. A butterfly-type inlet valve is nearly pressure-balanced and is utilized where positive shutoff is not a requirement.

2.1.3 Actuator

In direct-acting regulators, the sensor with its reference-load source generally is used to actuate the inlet valve. In dome-loaded units, a separate actuator is required; one type of actuator used in dome-loaded regulators is controlled by the output pressure of the loader valve as shown in figure 6. In a pilot-operated regulator, the actuator is positioned by the

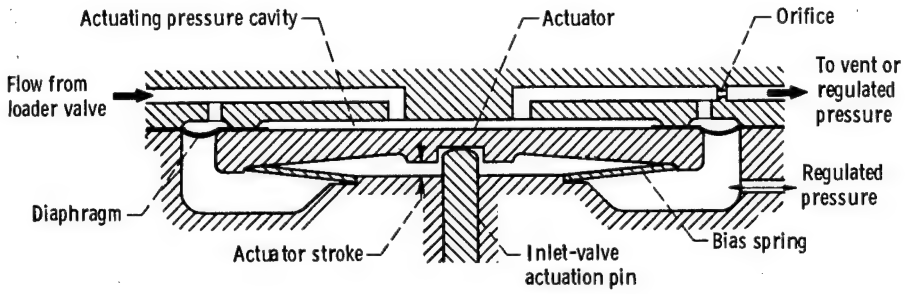


Figure 6. - Schematic of actuator for dome-loaded regulator.

actuating pressure resulting from the pilot valve or controller varying the rate of flow out of the actuating-pressure cavity into which a regulated-pressure flow is supplied, as shown in figure 7. Details on actuator designs, characteristics, materials, and associated problems are presented in reference 31.

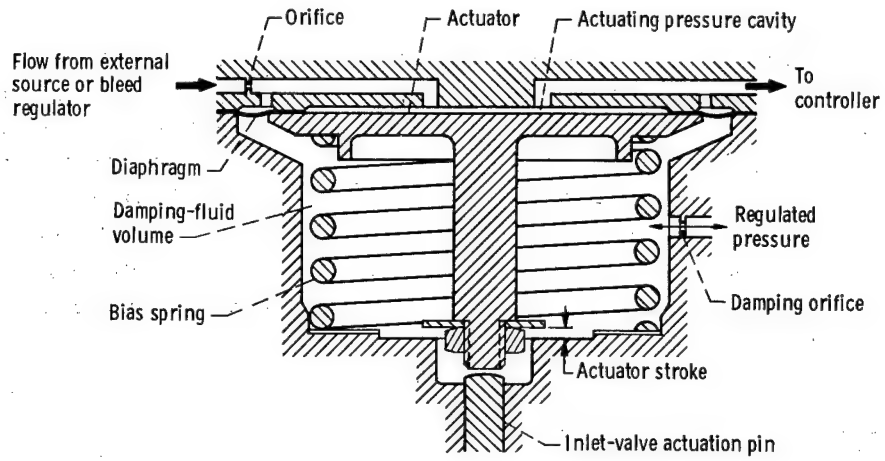


Figure 7. - Schematic of actuator for piloted regulator.

2.1.3.1 TYPE

The actuator may be a piston, diaphragm, bellows, or bladder; the choice depends on the application. Type of fluid, operating-temperature range, regulated-pressure tolerance, vibration and acceleration loads, response requirements, and stable operation must be evaluated in the design selection.

Pistons with O-ring dynamic seals may be used when temperature range and compatibility with the fluid permit the use of elastomers. Metallic diaphragms and bellows generally are used when the application involves exposure to extreme temperature or to corrosive fluids and when low friction is desired. Bellows of corrosion resistant steel (CRES) or beryllium copper are common; machined, welded, or brazed construction has been satisfactory. Diaphragms of Mylar, fiberglass cloth backed with Teflon, rubber-coated nylon cloth, and rubber have been used. The radius around which a diaphragm flexes is critical; normally this radius is a minimum of 0.01 in. (0.254 mm), and is designed to blend smoothly with the diaphragm clamping surface.

In Mylar actuator and sensor diaphragms, random failures attributable to rupture of the material have occurred; extensive testing simulating actual operating conditions with a variety of configurations has failed to indicate the basic cause. Mylar has mechanical properties that make it attractive for use as a diaphragm material: relatively high strength, ease of forming, low permeability factor, and a modulus that produces low spring rates; the advantages of this material, however, must be carefully evaluated against the consequences of random failure.

2.1.3.2 SIZE

The actuator size is a function of the regulated-pressure tolerance and the forces required to operate the inlet valve. To arrive at the proper size, the designer evaluates the errors caused by supply-pressure variation, inlet valve position, acceleration forces, mechanical hysteresis and friction, thermal effects, flow forces on valves, and vibration effects. For piloted regulators, the actuation-pressure variation is determined by the orifices controlling fluid flow and by the characteristics of the pilot valve or controller that senses the regulated-pressure variations. The gain of the overall system must be sufficient to result in the desired regulation. Common practice is to evaluate known variables and then provide additional actuating force for other second-order factors.

2.1.3.3 RESPONSE

Actuator response and stability of operation are functions of the relative volume of fluid surrounding the actuator and of the characteristics of the pilot valve or controller, the orifices that meter the flow of control fluid, and the bias spring. Two examples of operational problems and solutions illustrate these interrelationships:

(1) In a direct-acting regulator used on an Atlas engine, a sustained audible high-frequency instability was produced during high-flow operation. The volume of damping fluid was small, and the damping orifice was large (fig. 2). The damping-fluid volume and orifice size were changed to tune the circuit and eliminate the instability. A method of analyzing this type of design is given in references 32 (sec. 5.5), 33, and 34.

(2) In a piloted regulator used on a Saturn engine, the damping-fluid volume under the actuator was large and the damping orifice was small (fig. 7). During initial pressurization, the actuator response was slow; a high-pressure spike occurred in the regulated pressure, and the controller sensing diaphragm ruptured. The damping-fluid volume and orifice were changed to decrease the actuator response time and reduce the pressure spike to tolerable limits.

Variations in actuator stroke and friction and changes in the gas or mixture of gases also affect actuator response. A computer program with controlled arbitrary variations of these factors (ref. 21) is used to study the dynamic operation of the regulator.

2.1.3.4 BIAS SPRING

A bias spring is used with a dome-loaded or pilot-controlled regulator actuator to position the inlet valve to make the regulator normally open or normally closed. The actuators shown in figures 6 and 7 are opposed by bias springs; in each, the inlet valve is closed, and pressure from the loader valve or other source must exceed regulated pressure to open the

inlet valve. If the bias spring assisted the actuator, the inlet valve would be held open until the pressure from the loader valve increased to a level approaching regulated pressure. The characteristics of the bias spring thus can affect regulator operation.

In a reaction-control system, the propellant tank pressurization gas is supplied through a pressure-reducing regulator from bottles initially at a high pressure. As the gas in the bottles is used, the pressure at the regulator inlet decreases. The most effective use of the bottled gas is made when the regulator maintains the regulated pressure at minimum inlet pressure. In one instance, when a dome-loaded actuator of the type shown in figure 6 was operated with a minimum inlet pressure and high flow demand, the regulated pressure decreased below the required minimum. Then the actuation pressure supplied to the actuator by the loader valve was not sufficient to open the inlet valve fully. To compensate for this decrease in opening force, a variable-rate Belleville spring was used to reduce the closing force of the bias spring at the full-open position.

2.1.4 Control Elements

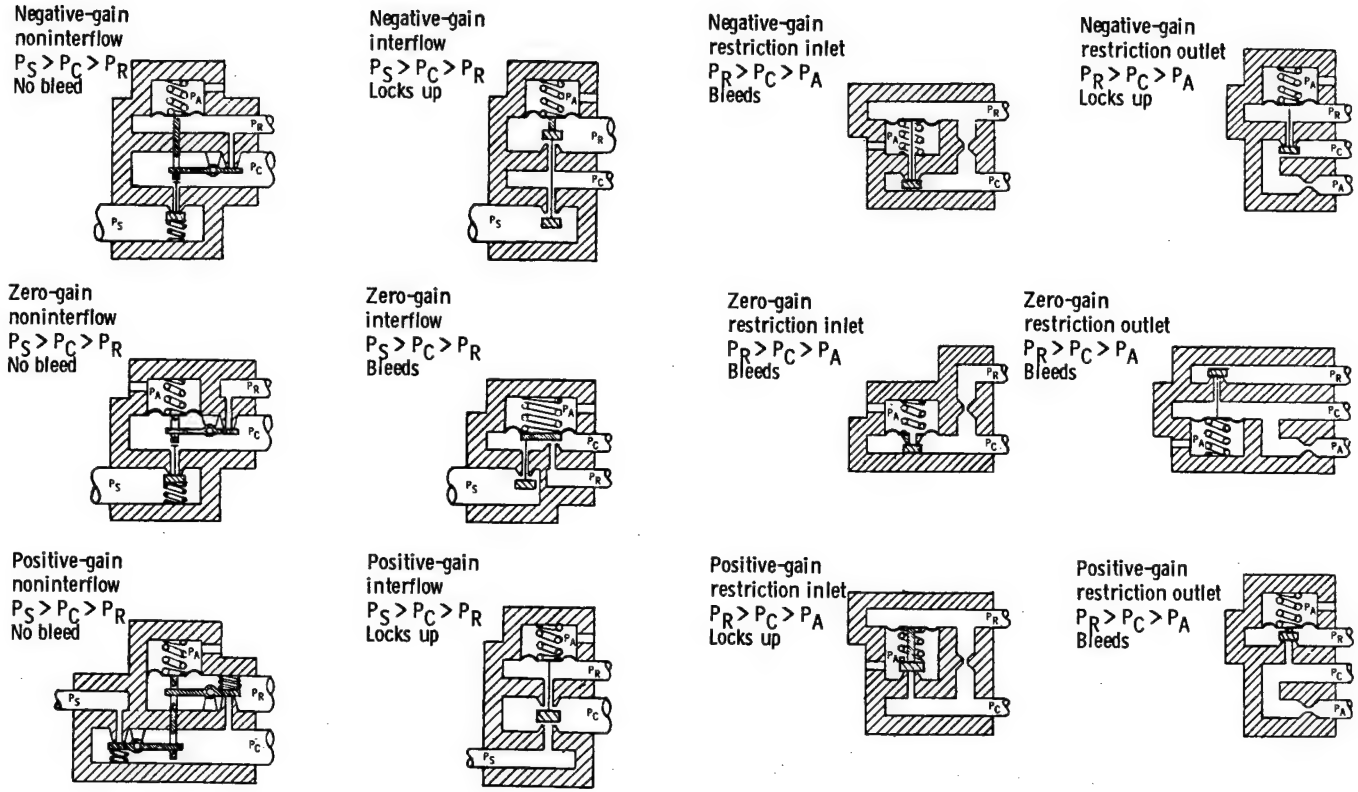
The regulator includes a variety of detail elements that are required to meet the specified performance. All regulators have a reference load that is adjustable to the regulated-pressure set point. Other control elements are pilot valves, controllers and bleed regulators, shutoff valves, thermal compensators, control orifices, and absolute-pressure reference.

2.1.4.1 SPRING REFERENCE LOAD

The reference load is normally a coil spring or Belleville spring. When an absolute-pressure reference is required, an evacuated bellows is used with or without a spring. The movements of the reference load and its moving parts are made as frictionless as possible because this source of regulated-pressure error may be amplified if there is gain in the system. The springs are supported in a manner that avoids introducing side loads. When ordinary springs create unacceptable side loads, springs with specially ground ends that are square with the axis when the spring is compressed to the operational length often are used (ref. 35). The theory of spring design is given in reference 36. Current practices for good spring design are discussed in reference 31. A method for stacking Belleville springs is given in reference 37.

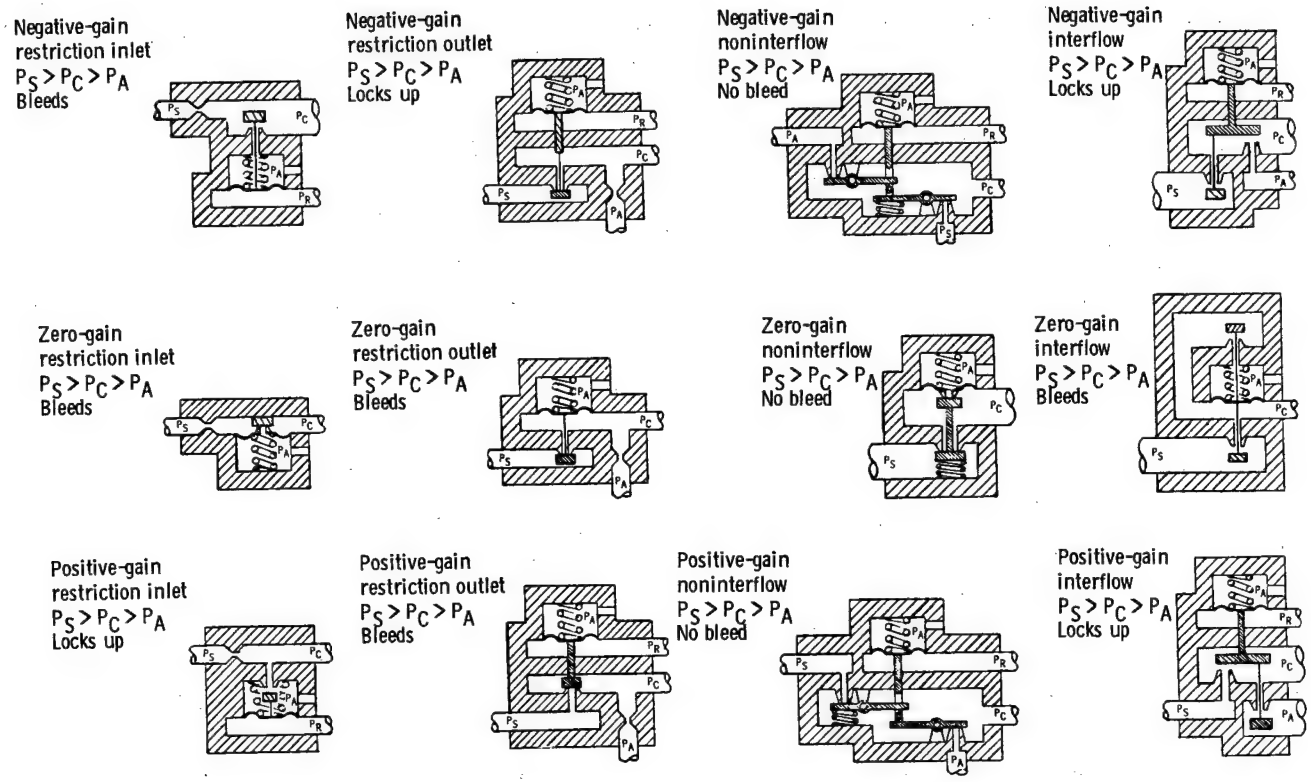
2.1.4.2 PILOT VALVE CIRCUIT

A variety of pilot circuits is available for use in piloted and dome-loaded regulators (ref. 38); these circuits are shown in figures 8 and 9. (All terms used in figures 8 and 9 are defined in the Glossary). At the start of a new design, the pilot circuits are reviewed and the most suitable type is selected.



(a) Exhaust of bleed flow to regulated pressure

Figure 8. - Pilot circuits with supply pressure as source of pilot pressure.



(b) Exhaust of bleed flow to atmosphere

Figure 8. - Pilot circuits with supply pressure as source of pilot pressure. Concluded.

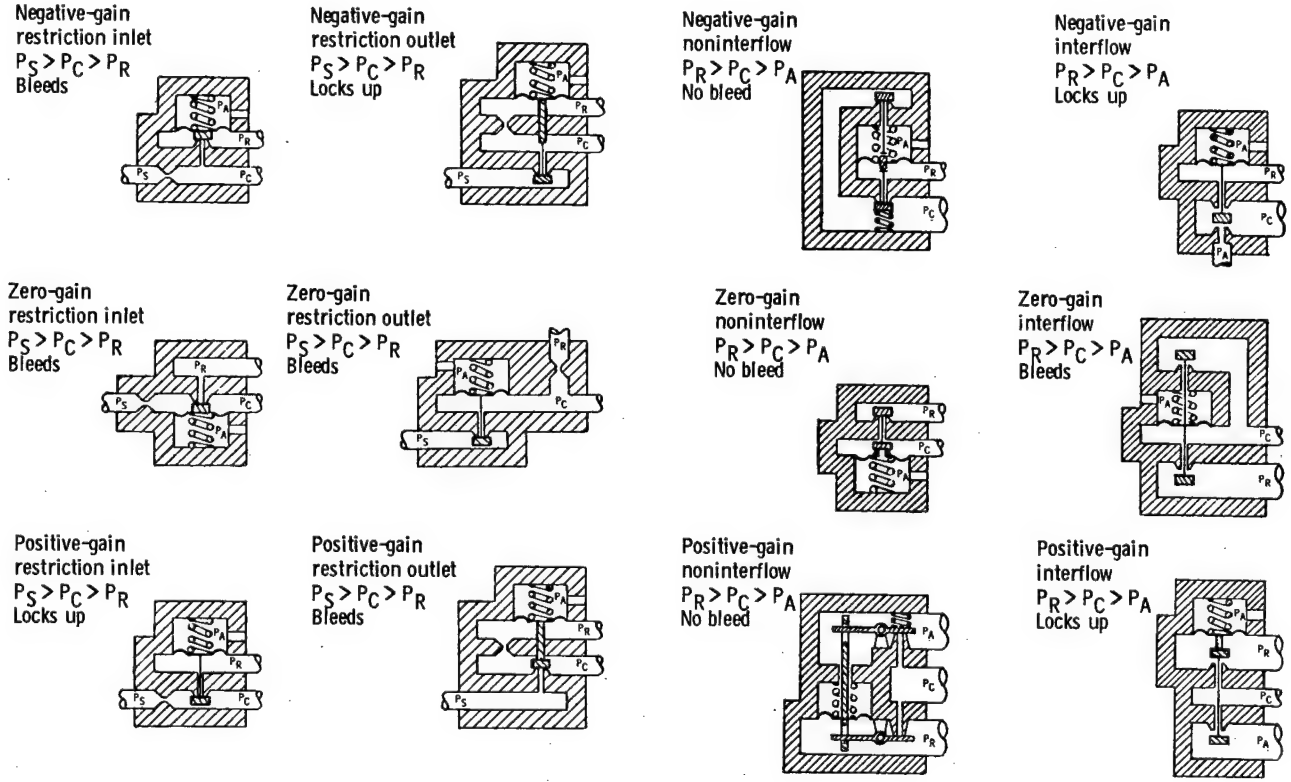


Figure 9. - Pilot circuits with regulated pressure as source of pilot pressure, exhaust of bleed flow to atmosphere.

The pilot circuits are divided into two classifications depending on the source of pressure to the pilot: (1) circuits using the supply pressure as a pressure source and (2) circuits using regulated pressure as a pressure source. Type (1) circuits may exhaust the pilot bleed flow to the regulated pressure (fig. 8(a)) or to atmosphere (fig. 8(b)); type (2) circuits must exhaust the pilot bleed flow to atmosphere (fig. 9). These latter circuits have limited application because of the exhaust-to-atmosphere requirement, but do have an advantage of a relatively constant source of pilot pressure.

Pilot circuits may have zero, negative, or positive gain. Zero-gain circuits generally are utilized on dome-loaded regulators, in which the pilot sensor references the inlet-valve-actuator control pressure and maintains the control pressure at the desired relatively constant pressure. In general, negative-gain pilot circuits are used for pressure-reducing regulators, and the positive-gain pilot circuits for back-pressure regulators. Noninterflow pilot circuits are seldom used, because of a possible regulated-pressure band in which the regulator does not control and the pressure may hunt between the pilot-valve limits. The interflow pilot circuit is complicated but has the advantage of reducing the recovery time when the regulated pressure overshoots or undershoots the allowable regulated-pressure range.

The location of the flow-path restriction as inlet or outlet affects the transient response of the regulator. With the restriction inlet, the regulator response time for a sudden termination of flow demand will be less than that for a sudden increase of flow demand. The restriction-outlet piloted regulator will have response characteristics opposite to those of the restriction inlet. The choice between a pilot circuit that will bleed or lock up, or will exhaust to atmosphere or to regulated pressure, depends on the requirements of the system controlled by the regulator.

2.1.4.3 ORIFICE

Orifices are used in regulators to control flow or to provide damping. For effective control by an orifice, the flow passages upstream and downstream of the orifice must be sufficiently large that the orifice is the primary restriction. When the allowable variation in controlled flow is nominal, the manufacture and inspection of sharp-edge orifices are controlled so that the orifice may be used as manufactured. When the control of flow must be precise, each orifice is flow-calibrated, and orifices not meeting calibration limits are reworked to specification. The installation of the orifice is controlled to ensure that the flow direction is correct. Orifice plates must be sealed adequately to prevent bypass flow. The control of flow by an orifice that is eccentric to the flow passage is described in reference 39; additional flow-control characteristics of orifices are given in references 40 through 44.

Sintered metal was used as a flow restrictor in early space vehicles. In cryogenic applications, frozen moisture ("snow") is trapped in the pores of the sintered metal. Complete failure of a regulator has occurred when the frozen moisture accumulated in the sintered material and blocked the flow of control gas.

2.1.4.4 THERMAL COMPENSATOR

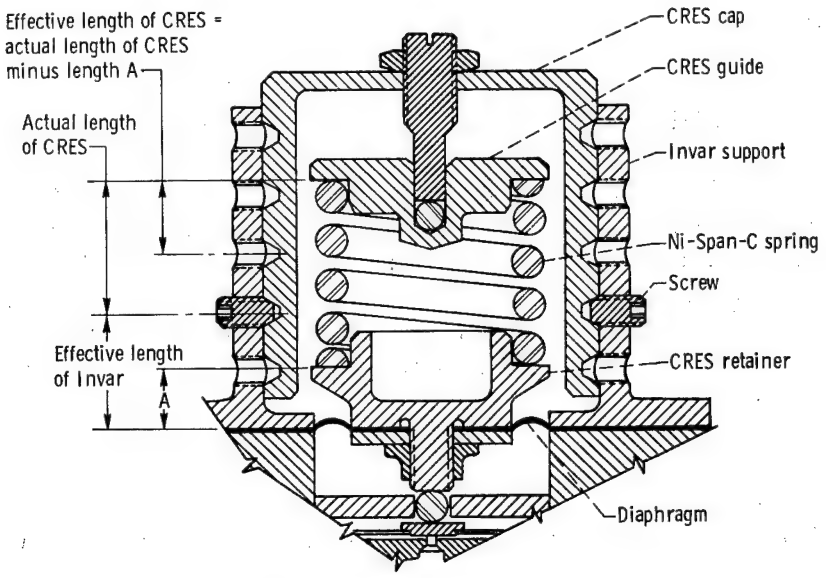
Accurate pressure regulation throughout a wide range of temperature requires provision for thermal compensation; uncompensated regulators will shift pressure setting approximately 3 percent per 100°F (55.6K) change in temperature. The reference load of the sensor is the critical item. When a coil spring is used, variations in the required reference load will occur because of thermal expansion or contraction of the supporting structure and changes in spring rate resulting from changes in modulus of rigidity (the major factor) and from thermal expansion or contraction (ref. 45). When a regulator is adjusted at room temperature and then cooled, the spring housing contracts, decreasing the installed length of the spring and thereby increasing the reference load; in addition, the cooling of the spring increases the spring rate. The resultant reference-spring load and regulator pressure setting is the product of effective increased deflection of the spring and the increased spring rate.

The spring-supporting structure in temperature-compensated designs uses combinations of materials to alter the installed spring length so that a constant reference load is maintained over the temperature range; Invar and CRES are preferred materials. The amount of compensation varies with the relative lengths of the elementary details. A means of varying the relative lengths of the metals is shown in figure 10(a); the location of the screws in the Invar governs the effective length of the two materials. With this type of compensator, it is possible to preclude almost completely a shift in regulated pressure setting as a result of temperature change. In an attempt to gain a greater difference in coefficients of thermal expansion, tests were made with Teflon, Kel-F, and nylon plugs; however, these materials tended to cold flow during long storage periods and did not return to their original dimensions when cycled through a wide variation in temperature.

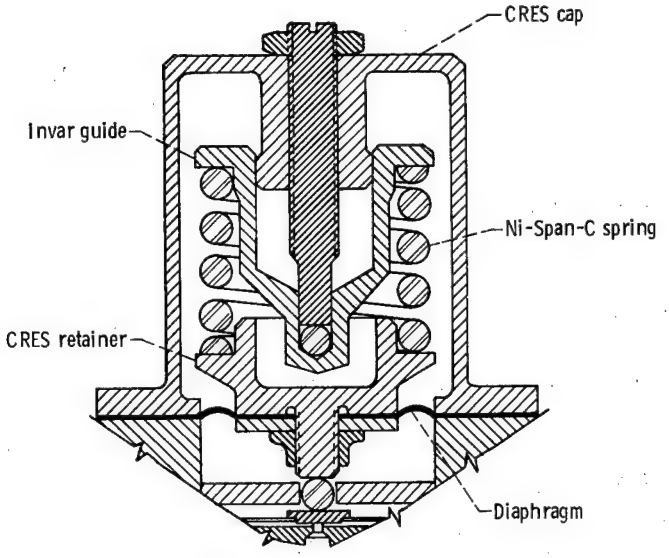
Another thermal compensation arrangement is shown in figure 10(b). To minimize the setting shift due to changes in spring rate, a spring material with the minimum change in thermal expansion and modulus of rigidity is used; Ni-Span-C wire is a common choice. With this type of thermal compensator the regulated-pressure setting will shift approximately 1 percent per 100°F (55.6K) change in temperature.

2.1.4.5 SHUTOFF VALVE

In some systems, a shutoff is required between the pressure source of bottled gas and the pressurized system. The shutoff valve normally is electrically operated and, if pressure-unbalanced-to-close for low leakage, will require a large, heavy actuator. To reduce the actuator size and weight, the valve may be pressure-balanced, with added complication and possible excessive leakage at high inlet pressures. With a normally closed regulator arrangement as shown in figure 4, the pressure-unbalanced solenoid shutoff valve may be used immediately downstream of the bleed-flow pressure regulator. In this location, the shutoff-valve flow capacity and inlet pressure are low, and the valve is small and light. The three leak paths – regulator inlet poppet, bleed-flow regulator poppet, and shutoff-valve



(a) Adjustable thermal compensator.



(b) Fixed thermal compensator.

Figure 10. - Types of thermal compensators.

poppet – involve pressure-unbalanced-to-close poppets that have low leakage. A low-capacity relief valve is added between the bleed-flow pressure regulator and the shutoff pilot valve to limit overpressurizing the low-pressure control circuit by leakage from the bleed-flow pressure regulator.

2.1.4.6 FLOW LIMITER

When a system having a small ullage and a normally open regulator is pressurized, the initial pressure transient can exceed the maximum allowable regulated pressure. This transient can be controlled or limited by restricting the flow to the regulator inlet. For example, in one reaction-control system, an initial pressure surge caused the relief valve to relieve and ruptured a burst disk. As the burst disk was installed in the system to limit external leakage during long hold periods, its function was destroyed. A flow limiter was added to the regulator inlet, and the regulator had time to close before an excess flow overpressurized the system.

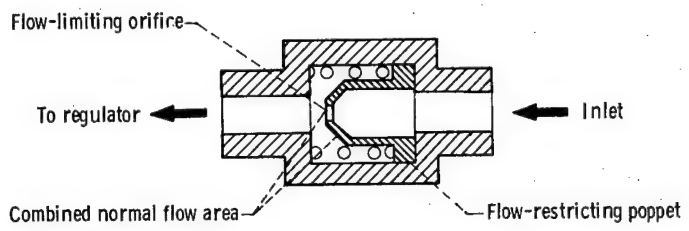
Two kinds of flow limiters are in use. In the step-type flow limiter shown in figure 11(a), the initial high flow creates a pressure drop across the flow-restricting poppet. This drop causes the poppet to move to the position where the flow-limiting orifice is the only flow passage. When the high flow stops, the poppet shuttles to the open position where flow area is sufficient for normal operation. A variable flow limiter, shown in figure 11(b), provides a movable piston that determines the open position of a metering valve; this type of unit can be designed to be effective over a range of the supply pressure. If either kind of flow limiter should fail in its actuated position, the normal function of the regulator will be affected. Because the flow limiter is a secondary device, the design must have high reliability.

2.1.5 Operational Problems

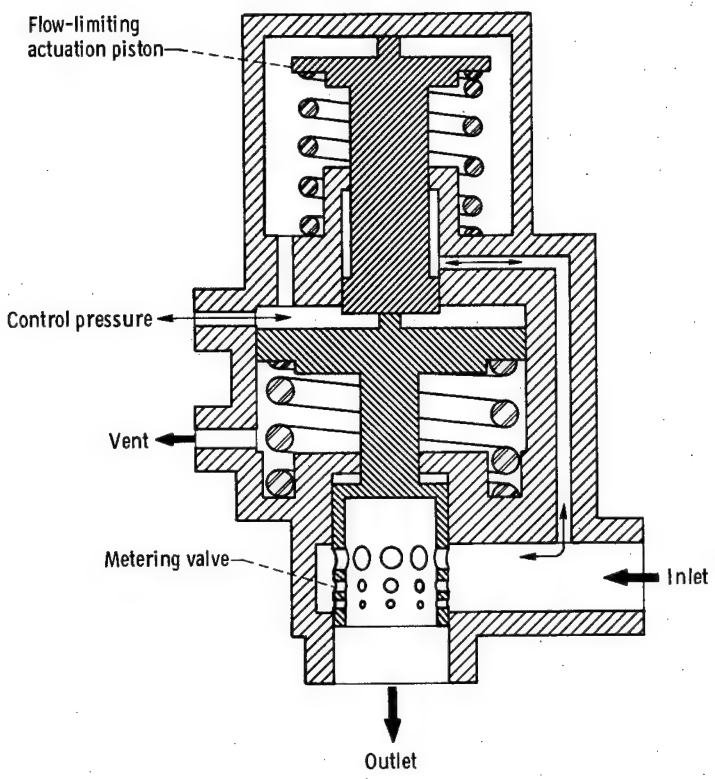
Regulators are subject to a variety of problems during operation. Redesign of the unit or careful control of operating procedures may be necessary. Typical problems involve contamination, lubricants, vibration, freezing of moisture in moving parts, and stability.

2.1.5.1 CONTAMINATION

The low-leakage requirements, critical flow passages, and close clearance of moving parts necessitate protecting the regulator from contaminated fluids. Metallic fragments from rupture disks and explosive valves have seriously damaged seats and working parts of regulators downstream from the disk or valve. In cryogenic applications, snow trapped in valve seats has caused severe leakage. When orifices have been plugged by solid particles, the regulator did not maintain the desired regulated pressure. When the regulated fluid is helium



(a) Step type.



(b) Variable type.

Figure 11. - Types of flow limiters.

or hydrogen, particles may be moving at the velocity of sound. Traveling at this high velocity, small particles have been known to puncture Mylar diaphragms; to prevent this, a filter is provided at the inlet of the regulator. The filter should have a low pressure drop and adequate contaminant handling capabilities; the pore size should be small enough to remove the size particle that may cause failure. Special filters may be added to protect critical orifices. Detailed information on filters is available in reference 46. Regulator internal details are not protected by the filter, however, and cleaning, handling, and assembly procedures must be established to prevent building contaminant into the assembly (ref. 47).

2.1.5.2 LUBRICANTS

In regulators with dynamic O-rings that are not lubricated by system fluids, a lubricant must be used to reduce the breakout and running friction. The lubricant must be effective over the operating temperature range and compatible with the fluid in the system being pressurized (e.g., oxygen, RP-1). Reference 48 provides a guide to the selection of lubricants. Regulators with lubricated O-rings have exhibited sustained pressure oscillation, transient operation beyond allowable regulated pressure limits, and failure to operate in cold environments. These malfunctions were caused, respectively, by excessive amounts of lubricant, by lubricant that had dried out and become sticky during long periods of storage, and by the use of an incorrect lubricant.

In cryogenic applications, O-rings are not used. A dry-film lubricant may be used to reduce friction and wear. This was the case in a regulator where a sliding push rod galled and the unit did not maintain allowable pressure regulation. The particles generated by the galling damaged other parts of the regulator. A dry-film lubricant was added to the push rod, and the failure was not repeated. The preferred dry-film lubricant is a type that is permanently bonded to the surfaces. Reference 49 provides guides for the selection of dry-film lubricants.

2.1.5.3 VIBRATION

In a piloted regulator with a continuous external bleed, the pilot valve controller varies the rate of fluid flow from the inlet valve actuator cavity. Because the flow into the actuating cavity is relatively constant, actuating pressure is a function of the exit flow through the controller. For a steady-state operating condition, the flow through the controller is constant, and the forces on the controller valve, sensor, and reference spring are balanced. In vibration environments, the spring-loaded mass of the controller resonates. The resonating controller valve varies the fluid flow from the actuator. Under these conditions, the regulated pressure oscillates or shifts from its normal setting and may exceed the permissible tolerance. With stops added to both ends of the controller valve travel, the flow variation is limited and the controller valve oscillates between full-open and full-closed. Since the nominal setting of the controller is mid-position for a steady-state condition, the controller-valve strokes for the fully open and fully closed time are approximately equal (average controller flow area unchanged), and the regulated pressure remains relatively

constant. Under transient flow conditions, the forces on the sensor are unbalanced, and the controller valve is either fully open or closed for a longer period of time. As the fluid flow through the controller valve changes, the pressure to the actuator changes and, as a result of a new inlet valve position, the regulated flow changes. At this new steady-state condition, the controller flow is constant, the controller-valve dwell-open period being longer or shorter because of the new balanced-force condition of the controller. The shift in regulated pressure producing the new balanced-force condition of the controller is the regulated-pressure error due to the new flow demand. A vibration-induced resonating condition in a controller valve with stops thus limiting the poppet stroke as necessary will have a minimal effect on regulated-pressure error. Shock mounting has been utilized to eliminate induced vibration; however, this method generally is ineffective when metallic lines are used or when cryogenic fluids preclude the use of elastomeric compounds.

2.1.5.4 MOISTURE IN REFERENCE CAVITY

In most regulators, the sensors are exposed to the regulated pressure on the inner surface and must reference atmospheric pressure on the outer surface. The atmospheric reference normally contains the reference spring in a retaining closure or spring cavity with an opening to atmosphere to permit breathing due to sensor motion. The spring cavity must be protected from liquids, moist air, or other foreign matter that can enter through the breather opening. Frozen liquids have stopped sensor stroking, closed motion clearances, altered spring characteristics, damaged diaphragms, and resulted in regulator malfunction. When regulators are used to blow down tanks, the regulator becomes chilled due to the thermodynamic processes involved, and moisture condenses in the reference-spring cavity. This condensed moisture must be removed between blowdown tests to avoid failures that can occur if the moisture freezes, or the moisture must be prevented from reaching the cavity.

One solution to the moisture problem is to provide a breather tube routed so that it will restrict entrance of liquids into the spring cavity; however, the tube diameter must be sufficiently large to prevent capillary action from drawing liquid into the cavity.

The cavity can be protected from the entrance of moisture by covering it. Covers of different shapes may be used. The liquid deflector with multiple noninterconnecting airflow passages shown in figure 12 has prevented liquids from entering spring cavities while the regulator was exposed to simulated or actual rain for extended periods of time.

Another solution to the problem of preventing moisture in the reference cavity is the use of a vent port check valve (sec. 2.3.3.5). However, the valve operation creates a variable back pressure on the sensor, and these valves are not used when accurate regulation is required.

In general, however, regardless of the method used to protect the reference cavity, the regulator moving parts that may be exposed to moisture or other contamination are less susceptible to failure when the designer avoids close clearances with long overlaps.

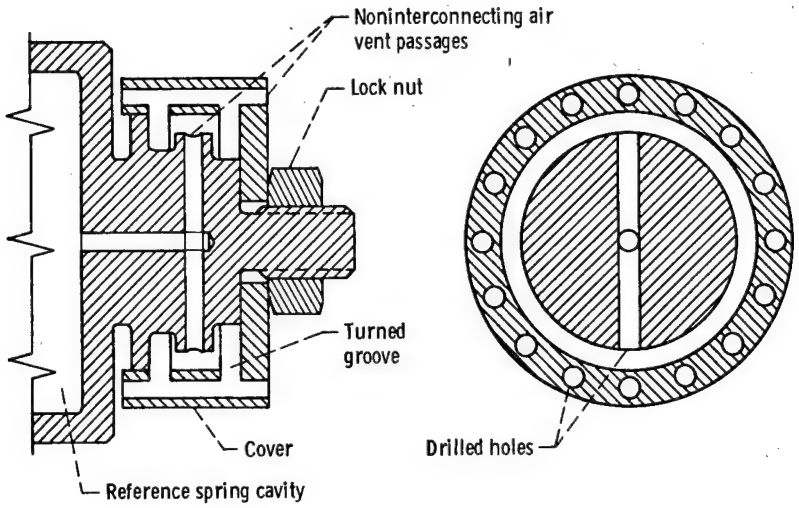


Figure 12. - Cover for atmospheric-pressure reference cavity cover.

2.1.5.5 STABILITY

Under actual operating conditions, a regulator can become unstable if flow dynamics reduce to a relatively low level the forces available for damping. Location of the sensing passage from the flowing section to the controller section is important for stability. If the sense point is in an area of flow turbulence, unstable operation can result. If the sense point is located in a region where cross-sectional flow area changes, an erroneous outlet pressure can result. Remote sensing, i.e., sense point is located a significant distance downstream of the regulator, introduces an inherent instability created by transmission lag and pressure drop between the regulator and the sense point. Sustained flow rates higher than normal can also result in regulator instability. For example, a direct-operated regulator pressurizing an engine lubricating-oil tank performed satisfactorily under normal flow conditions. However, when an empty tank was pressurized during engine checkout, the sustained abnormally high gas flow produced oscillations that resulted in severe damage to the regulator actuator mechanism. Considerable redesign effort was expended in making the regulator acceptable for the checkout condition.

Regulators exhibiting extremely high gain (i.e., a small error signal results in a large corrective action) are often unstable and require additional analysis to ensure stability. The piloted closed-loop inlet valve described in section 2.1.2.3 is an example of a basically unstable high-gain device. The inlet valve was utilized in an advanced-development regulator (ref. 19). A digital-computer program for analysis of the regulator had indicated that the inlet valve would limit cycle. Because of the low amplitude and high frequency of the limit cycling, there was no indication that regulated pressure would also limit cycle. When the

regulator was tested, the significance of the limit cycling became apparent when the inlet valve was destroyed by unstable operation. Coulomb-damping rings were added as a method of stabilization. It should be noted that the digital computer program used for the analysis was relatively unsophisticated in comparison with the capabilities of currently available analyses (ref. 21). The program synthesized only the regulator, whereas present practice is to synthesize the entire system so that the interaction of all components can be studied.

2.1.6 Leak Testing

When pneumatic regulators are required to lock up, it is desirable to determine the inlet valve seat leakage in the lockup condition. Leakage testing methods for general use are described in reference 50. In some missions, however, the vehicle may coast in space for long periods of time, and leakage by the locked-up regulator functioning as a shutoff valve could deplete the supply of bottled gas and result in mission failure. Further, the regulator lockup leakage discharging into a closed system can cause an increase in initial actuation pressure that changes valve operating times; normal engine starting can be affected by a change in valve operating times. Most regulator functional tests check the inlet valve leakage with supply pressure at the inlet and atmospheric pressure at the outlet. If the allowable leakage is less than 10 cm^3/hr, flowmeters cannot be used. With supply pressure upstream and regulated pressure downstream of the inlet valve, halogen or helium leak detectors cannot be used. Thus, a special leak test (described fully in section 3.1.6), is used to verify that the maximum allowable leakage will not be exceeded with the regulator in the lockup condition.

The special lockup leak test is based on measured changes in pressure and temperature of the regulated pressure of a locked-up regulator over an extended period of time. The regulator outlet is connected to a known volume. With a constant supply pressure, the changes in pressure and temperature for a time period are noted. These values are substituted in equation (4) (sec. 3.1.6); the solution indicates the leakage. The equation includes terms for the gas characteristics and the system physical variables. The numerical values of the variables are chosen to reflect the most adverse conditions. Therefore, the leakage calculated will never be less than actual leakage.

2.2 RELIEF VALVES

Relief valves protect fluid-system components from overpressurization. Overpressurization typically results from thermal changes or from leaking valves. For low flows, simple, direct-operating relief valves are used; for high flows such as those on large-vehicle tanks, pilot-operated relief units are used. Relief valves differ from most valves in that their operation creates a high pressure drop during flow; other valves are designed for minimal pressure drop.

To select a configuration, a tradeoff study to evaluate all the design requirements is made. Sufficient analysis is performed to provide a basis for comparing the operational characteristics of the various candidate arrangements. The chief factors considered include flow, full-flow and reseat pressures, temperature range, response, vibration, acceleration, overrides, possible dual-pilot operation, and the location of the unit in the vehicle. In some cases, the valve may be required to relieve a mixture of gases or propellant vapors. The detail design must consider allowable leakage, use of dynamic seals, and selection of materials compatible with the fluids handled. When performance is equal, the choice between alternate arrangements is determined by secondary features such as number of leak paths, ease of assembly, adjustment of set point, locking devices, size, weight, maintainability, reliability, and cost.

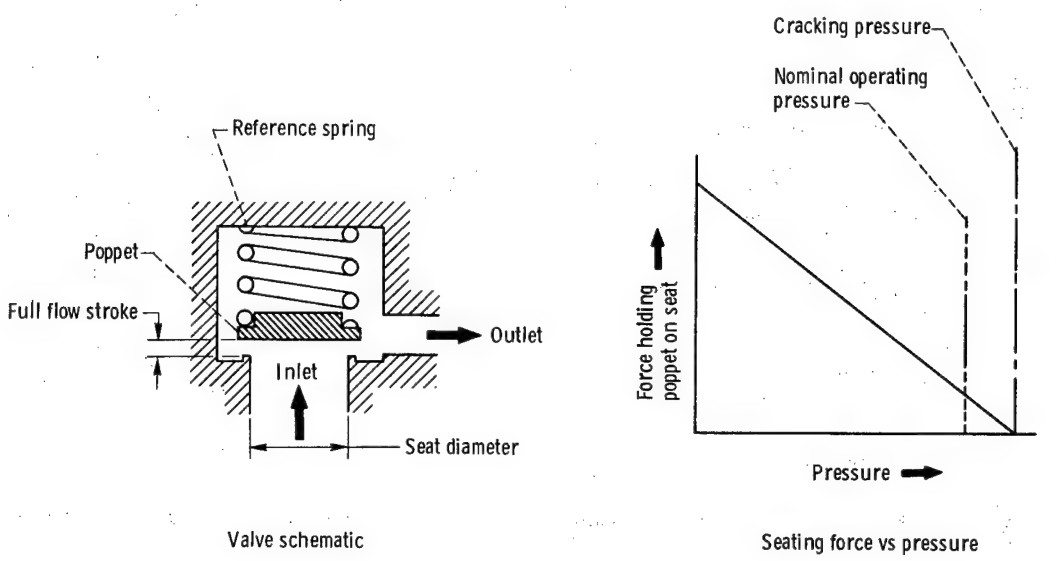
A specialized version of the relief valve is used on vehicle tanks. During the filling of a tank, the main poppet of the relief valve is held open by an actuator and the unit acts as a vent valve with a low pressure drop. A device that performs both of these functions is called a vent-and-relief valve. These units may have position indicators, solenoid valves for controlling the venting actuator, and dual pilots: one for normal operation, and the other for proof-testing the vehicle tank. The "normal" pilot is isolated during use of the "test" pilot. The design must be reviewed carefully to ensure that the normal relieving function of the valve cannot be blocked inadvertently.

2.2.1 Size

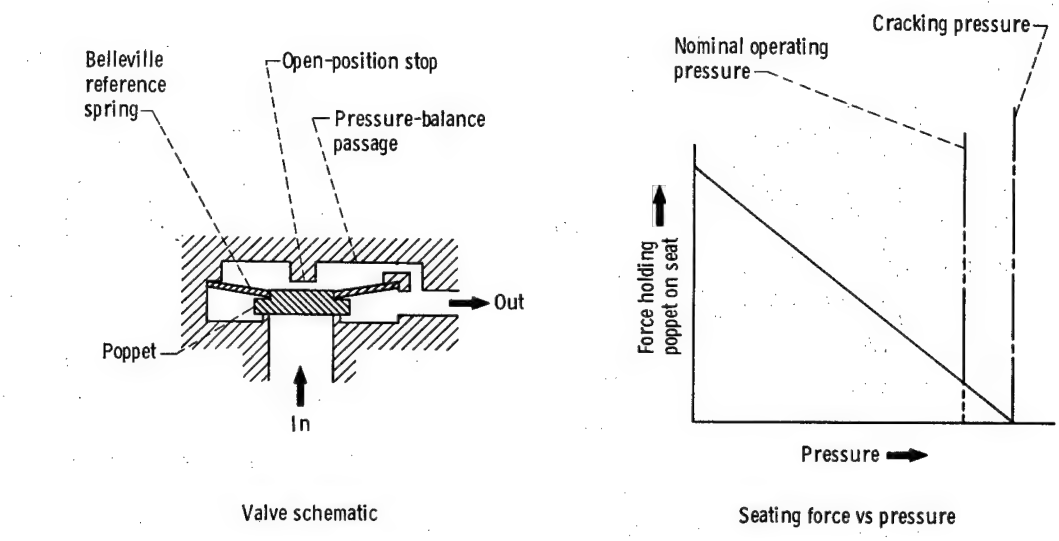
The first step in choosing a relief valve is to determine the area necessary to handle the required flow. The relieving flow area is determined for the maximum flow of gas at maximum allowable relieving pressure. The equation for isentropic compressible flow through orifices (ref. 25) is used in this calculation. The relieving flow area must be adequate to prevent system overpressurization under all operating conditions. If the relief valve is protecting a system with a regulator, the relief valve must be capable of flowing the discharge of a regulator that has failed in the fully open position (a "runaway" regulator).

2.2.2 Assembly

Relief valves may be direct-operating, inverted-operation, or pilot-operated. The direct-operating and inverted-operation relief valves are basic arrangements; typical schematics are shown in figures 13 and 14. The accompanying graphs of seating force vs pressure indicate the anticipated leakage characteristics of these arrangements. As leakage tends to be a function of the force holding the poppet on the seat, the seating force shown is the force resulting from the application of the reference-spring closing force and a varying opening or closing force due to the increasing system pressure acting on the pressure-unbalanced area of the poppet. At cracking pressure, the force holding the poppet on the seat is zero.



(a) Modulating.



(b) Nonmodulating.

Figure 13. - Valve schematic and seating force-vs-pressure plot for direct-operating relief valves (pressure unbalanced to open).

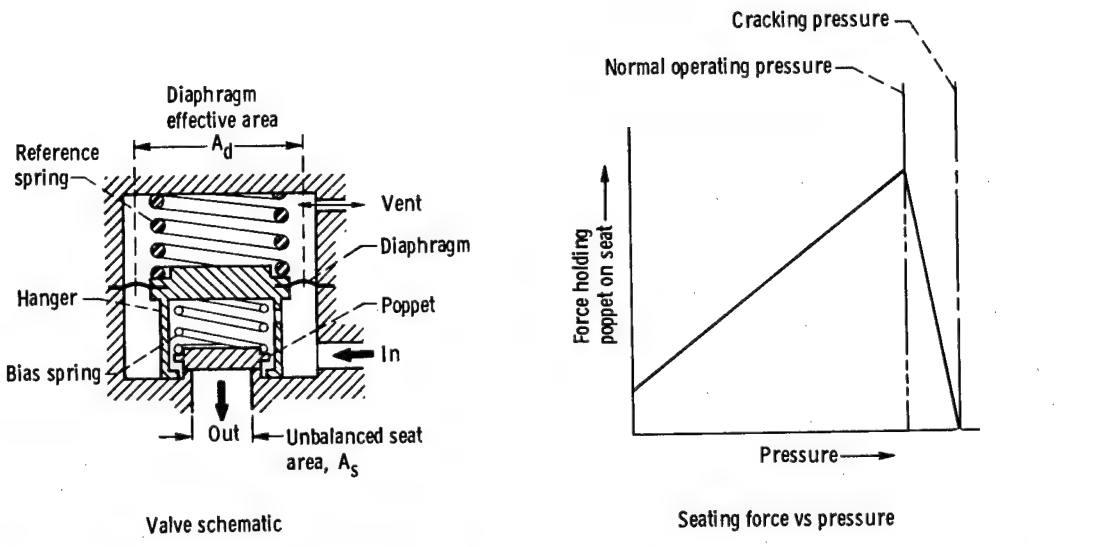


Figure 14. - Valve schematic and seating force-vs-pressure plot for inverted-operation relief valve (pressure unbalanced to close).

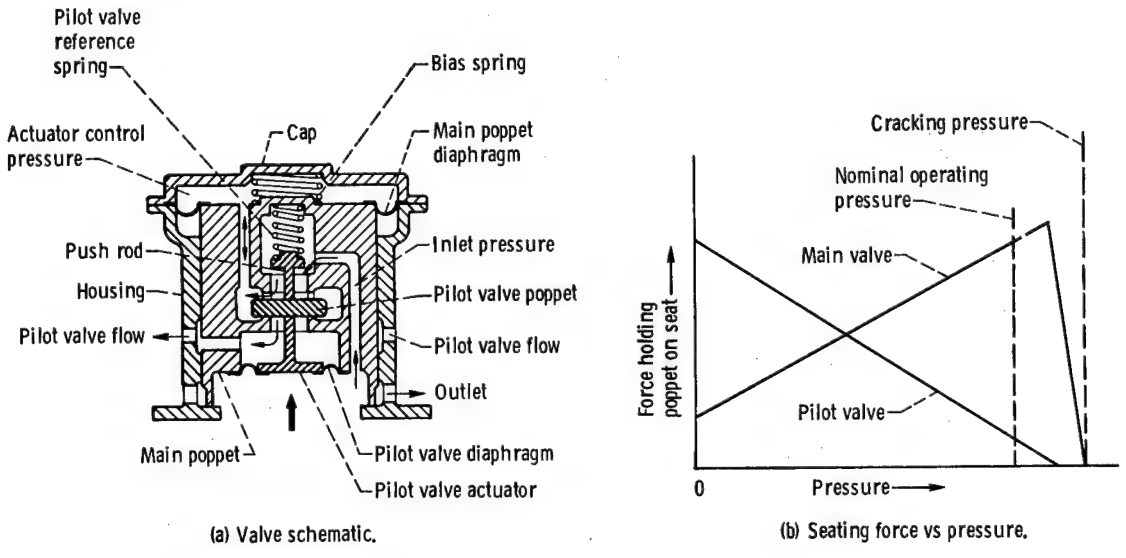


Figure 15. - Valve schematic and seating force-vs-pressure plot for pilot-operated relief valve.

The pilot-operated assembly (fig. 15) is a combination of the direct-operating and the inverted-operation basic arrangements. Additional control features are added to the large-tank relief valves as described previously.

2.2.2.1 DIRECT OPERATING

In the direct-operating relief valve shown in figure 13(a), the reference-spring load is applied at all times. The force of the poppet on the seat is the difference between the spring closing force and the pressure opening force. At the cracking pressure[1], these forces are balanced, and the poppet seating force is zero. As shown, the force of the poppet on the seat decreases with increasing pressure, and at nominal operating pressure a relatively small force is available for sealing against leakage.

The values for relief valve seat diameter and poppet stroke are established so that the flow area is equal to the required relieving flow area. Since an infinite number of combinations of seat diameter and poppet stroke exists, a method for selecting the best combination is needed. The graph shown in figure 16 illustrates the variation of relief valve parameters as a

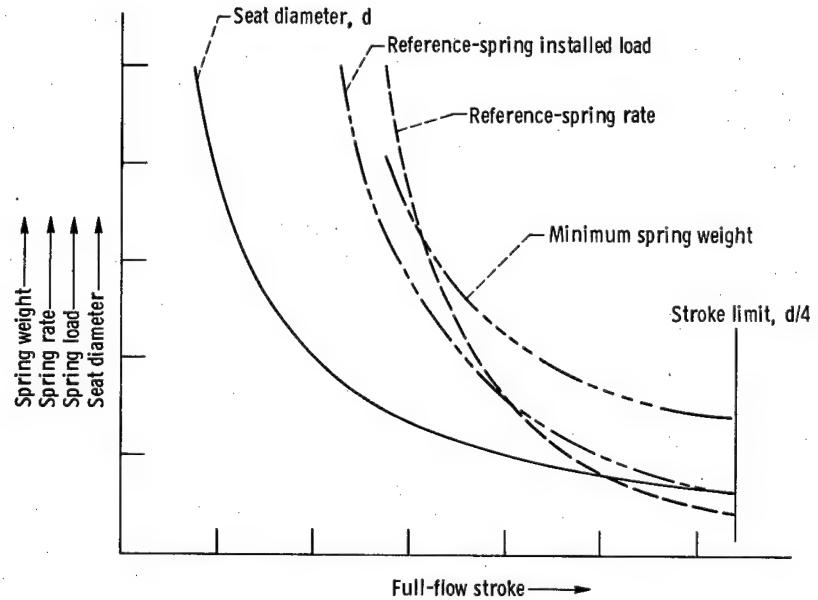


Figure 16. - Variation of relief valve parameters as a function of poppet stroke (direct-operating valve).

function of poppet stroke for full-flow conditions. As the full-flow stroke increases, the required seat diameter decreases. Since the reference-spring installed load is a function of the cracking pressure (a constant) and the seat diameter, the reference-spring installed load also will decrease as the full-flow stroke increases. The reference-spring rate is a function of the pressure differential between cracking and full-flow pressure (a constant), the seat

[1]Theoretically, cracking and reseat pressure are identical. In actual practice, reseat pressure usually is defined as the pressure at which a maximum allowable leakage will not be exceeded. This pressure usually is the upper limit of the nominal operating pressure.

diameter (varies directly), and the full-flow stroke (varies indirectly); thus the reference-spring rate decreases with full-flow stroke. If the reference-spring load and rate for any given full-flow stroke are known, it is possible to select a spring for that condition that will have a minimum weight. The minimum spring weight decreases with increasing full-flow stroke. The curves in figure 16 thus indicate that, within the full-flow stroke limitations, the seat diameter should be as small as possible. For maximum effective area, the full-flow stroke for a flat-poppet-and-seat configuration should not exceed one-fourth the seat diameter.

A variation of the direct-operating unit is shown in figure 13(b). In this unit, the reference spring is a Belleville spring with load-deflection characteristics that include a decreasing spring force with increasing stroke. The Belleville spring is installed to operate in this negative-rate portion of the load-deflection curve. The seating-force/pressure characteristics from zero to the cracking pressure are the same as those shown in figure 13(a). With additional increase in pressure, the poppet starts to stroke, the closing force of the reference spring decreases, and the poppet travels to the open-position stop. The poppet will remain on the open-position stop until the pressure decreases below cracking pressure and then will stroke to the closed position. This type of unit will not modulate; it has been used where excessive leakage is induced by vibration and where the relieving flowrate is low and a single Belleville spring can supply the reference force.

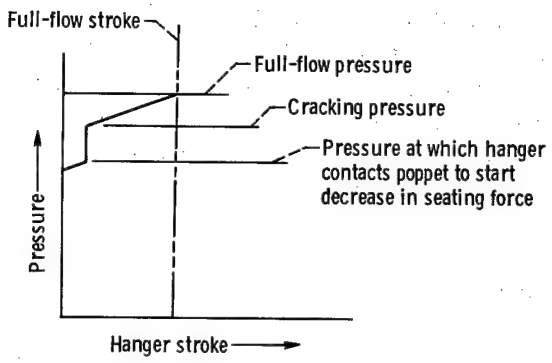
For high flows and high pressures, the helical reference spring of the direct-operating units becomes too large and heavy for use in space vehicles. Stacked Belleville springs are used when the hysteresis does not result in operation outside of the acceptable relieving-pressure range.

2.2.2.2 INVERTED OPERATION

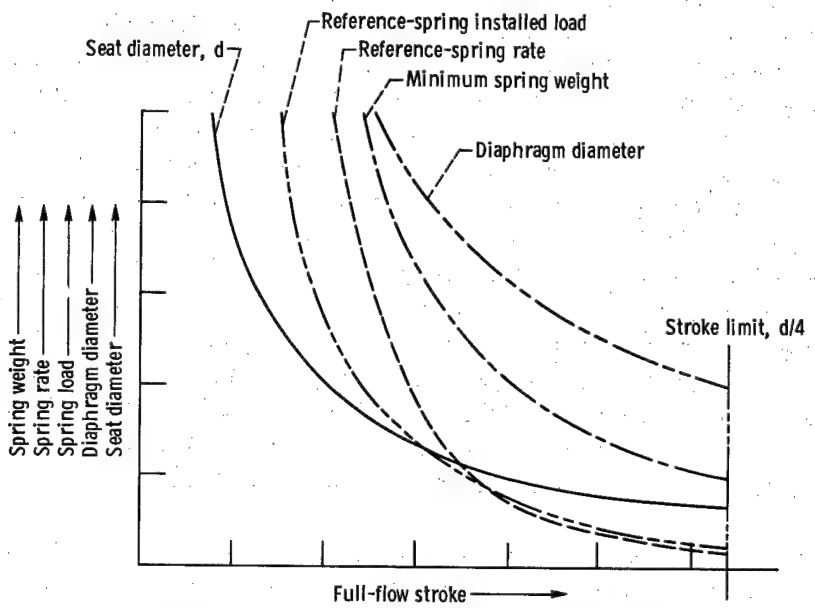
In the inverted-operation relief unit, the force holding the poppet on the seat increases with increasing pressure (fig. 14). It is possible to size the various elements of the unit to achieve maximum poppet-on-seat force at nominal operating pressure. The diaphragm senses the increasing pressure and at some pressure level will stroke the hanger that contacts the poppet and reduces the closing force. When the forces applied to the poppet are equal, the valve is at the cracking pressure. The bias spring applies a load to the poppet for initial positioning and low-pressure sealing. Figure 17(a) illustrates the relationship of hanger stroke to valve operation as inlet pressure varies.

For a maximum force to hold the poppet on the seat at a specified pressure P_t below the cracking pressure P_{cr}, the diaphragm effective area A_d is larger than the unbalanced seat area A_s by a factor given by

$$A_d \cong \left[\frac{P_{cr}}{P_{cr}-P_t}\right] A_s \qquad (1)$$



(a) Inlet pressure vs hanger stroke.



(b) Seat, spring, and diaphragm parameters vs full-flow stroke.

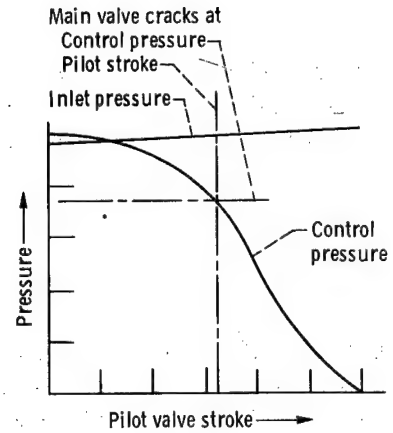
Figure 17. - Operating characteristics of inverted-operation relief valve.

The diaphragm effective diameter will be increasingly larger than the seat diameter as the maximum poppet-to-seat force occurs at a pressure closer to the cracking pressure. The graph in figure 17(b) illustrates variation of the inverted-operation relief valve parameters as a function of poppet stroke for full-flow conditions. These curves are based on a maximum poppet-to-seat force occurring at a pressure equal to 90 percent of cracking pressure. The characteristics of these curves are similar to those for the direct-acting relief valve, and the same selection criteria will be effective. The inverted-operation relief valve requies a greater number of parts and a more complicated construction. Each application should be evaluated to determine whether the improved leakage characteristics justify the additional parts and cost of the indirect-operating type.

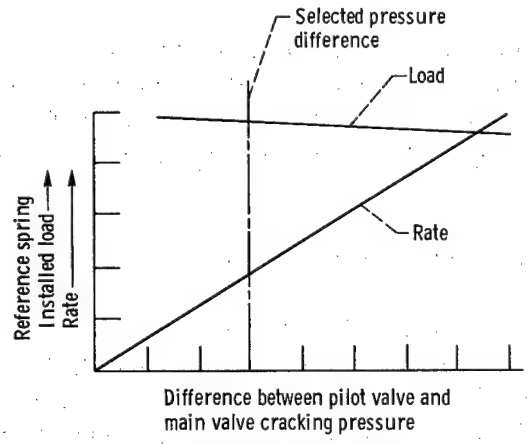
2.2.2.3 PILOT OPERATED

Pilot-operated units normally are used in systems where it may be necessary to relieve large flows. A typical pilot-operated relief valve is shown in figure 15(a). The main valve has the poppet-to-seat force-vs-pressure characteristics of inverted operation as shown in figure 14(b); to minimize leakage, the maximum poppet-to-seat force occurs at approximately normal operating pressure. The pilot valve has the poppet-to-seat force-vs-pressure characteristics of direct operation (fig. 13(a)). In the nonrelieving condition, the actuator control pressure is at inlet pressure, and the force resulting from the main-poppet pressure/area relationship holds the main poppet closed. When inlet pressure increases to cracking pressure, the pilot valve partially strokes, the actuator control pressure is ported to the outlet cavity, the pressure in the actuator control-pressure cavity is reduced to the extent that the force resulting from main-poppet pressure/area relationship is zero, and the main poppet is ready to open. As inlet pressure continues to increase, the pilot valve stroke increases, the pressure in the control-pressure cavity decreases, and the resulting force opens the main poppet. The amount the main poppet strokes is dependent upon the pilot valve stroke, which is proportional to inlet pressure. The response of the valve to changes in inlet pressure is a function of pilot valve flow capacity, actuator control-pressure-cavity volume, pilot valve sensing, characteristics of the reference spring, and characteristics of the fluid. If the relief valve has remote sensing, the size, length, and configuration of the sensing line is critical. In cryogenic applications, the possibility of mixed-phase flow in the sensing line is avoided by locating the sensing point in the tank where the ullage does not contain liquids.

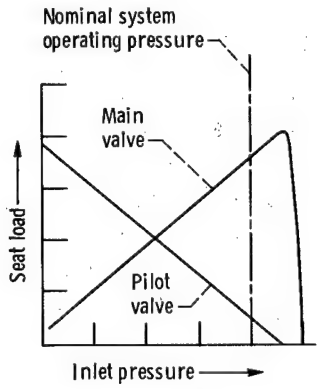
The curves shown in figure 18 provide a basis for evaluating a piloted relief valve configuration. Figure 18(a) shows the variation of the control pressure with pilot valve stroke as inlet pressure increases; the pilot stroke at which the control pressure equals the valve cracking pressure is thus established. Figure 18(b) indicates the reference-spring load and rate as the pressure difference between pilot valve cracking pressure and main valve cracking pressure varies. The curve in figure 18(c) gives an indication of possible valve leakage as the poppet-to-seat load of the pilot valve decreases with increasing inlet pressure and reduces the force resisting leakages. In a large valve, this leakage will be of secondary



(a) Pilot valve stroke vs inlet and control pressure.



(b) Reference-spring load and rate variation with cracking pressure difference.



(c) Poppet-to-seat load variation with inlet pressure.

Figure 18. - Operating characteristics of pilot-operated relief valve.

importance as the main poppet that is the source of large leakage has a large poppet-to-seat force. A steady-state analysis is conducted to determine the size and characteristics of the valve (refs. 51 and 52).

2.2.3 Inlet Valve

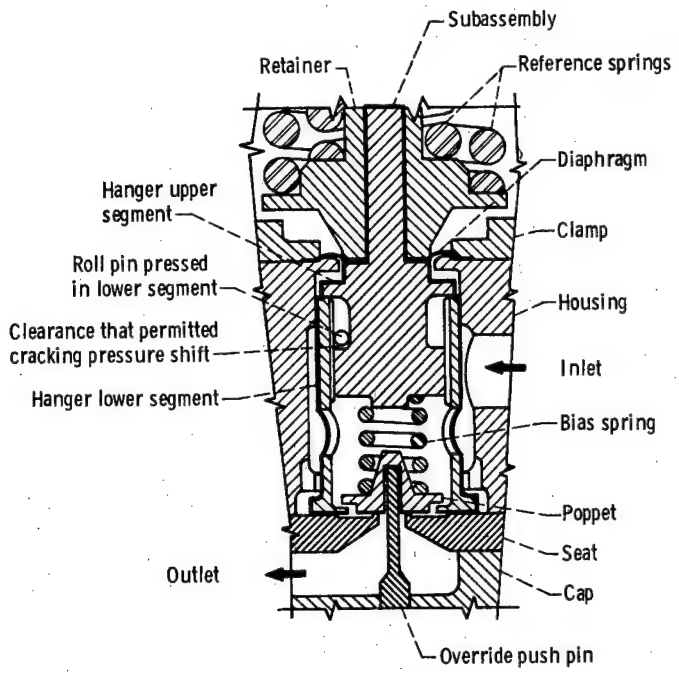
The inlet valve is the primary relieving flow path of the relief valve. As the relief valve is a high-pressure-drop device, the inlet valve design is selected for its sealing capabilities and repeatability of its cracking and reseat characteristics. These attributes are functions of poppet and seat design and, to some extent, subassembly construction.

2.2.3.1 POPPET AND SEAT

The primary problem in the design of the inlet valve is achieving low leakage without affecting the relieving-pressure repeatability. In direct- or inverted-operation units or pilot valves of pilot-operated relief valves, a change in the poppet-to-seat effective area will cause a shift in the pressure setting. If the relieving pressure shifts downward, the relief valve may be open in the normal range of regulator outlet pressure; if so, the stored gas intended to power a control system would be depleted, and the entire system would fail. During cycling, soft materials used for either the poppet or seat or both will wear, and the effective area will change. (The main poppet or seat of pilot-operated units may use soft materials, such as Kel-F lip seals, since a change in the effective area will not result in a shift of the pressure setting.) Leakage is a direct function of the seat loading stress, the surface finish (assuming a continuous contact between the poppet and seat over the sealing surface), and the fluid. For an applied load and seat diameter, the seat stress is controlled by the sealing surface width (ref. 53). Sealing surface widths and finishes can be controlled and inspected for various configurations such as ball and cone-shaped poppet on sharp-corner seats and flat poppets on narrow annular raised seats; the latter combination generally is less expensive to manufacture and inspect. Additional details on the design of poppets and seats can be found in reference 26.

2.2.3.2 SUBASSEMBLY CONSTRUCTION

A subassembly is an assembly of parts that operates as a unit in the valve. In the design and construction of a subassembly, special care must be directed to preventing any component from unlocking, disengaging, or shifting position relative to the other parts. For example, in one application of a design using the inverted-operation, pressure-unbalanced-to-close arrangement shown in figure 14, the hanger stroked with increasing pressure and lifted the poppet off the seat. The hanger had been constructed in two pieces so that the spring and poppet could be assembled within the hanger (fig. 19(a)); a threaded connection joined the



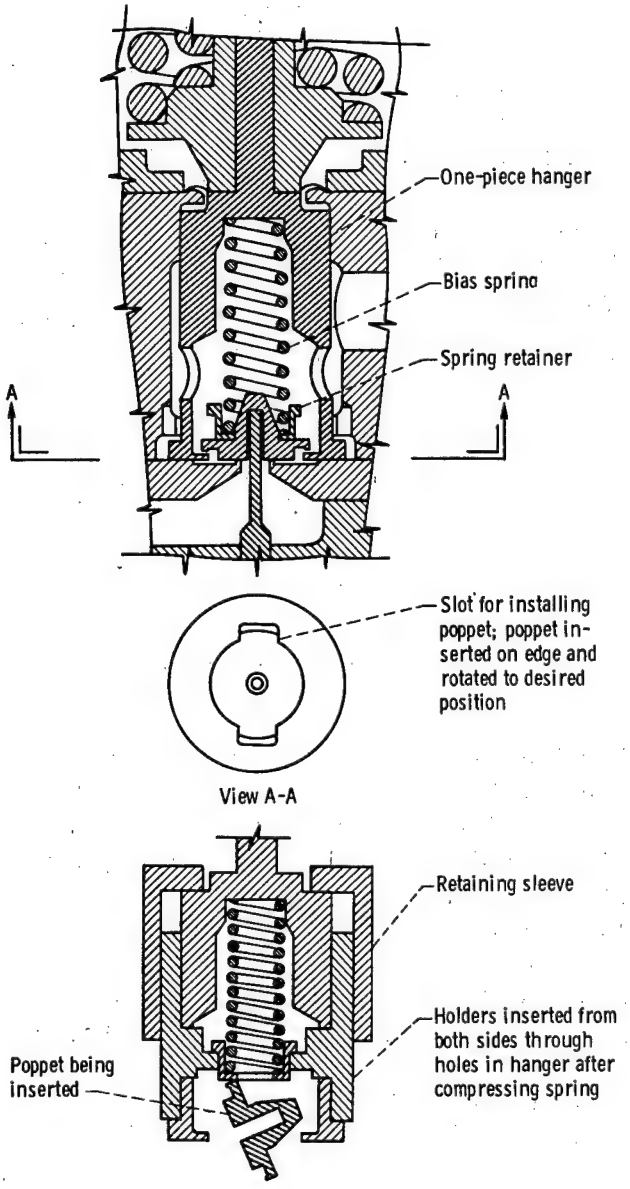
(a) Failed construction.

Figure 19. - Inlet valve subassembly construction.

two pieces. During vibration, the two pieces unscrewed, and the cracking pressure of the valve increased. The amount the pieces could unscrew and thereby increase the cracking pressure was limited by a pin pressed in one part that extended into a turned groove in the other part. However, even this limited shift in relief pressure was not acceptable, because the resulting higher gas pressure in the engine turbine-spin bottle would result in a high-energy engine start and possible engine malfunction. A positive locking method was required. Nylon-pellet locks could not be used because interference was lost as a result of differential contraction at cryogenic temperatures; pressing a pin through a hole drilled through the threaded area after assembly made disassembly and reassembly difficult. A single-piece construction was devised in which the poppet was inserted in a slot in the lower portion of the hanger, rotated into position, and held in place with the bias spring (fig. 19(b)).

2.2.4 Actuator

Actuators in relief valve assemblies position the main inlet valve or the pilot valve poppet. The actuator types in use and the practices for the selection of actuators for relief valves are the same as discussed in section 2.1.3 for regulator actuators. The problems with regulators described in sections 2.1.5.2 and 2.1.5.4 also have occurred in relief valve operation.


One-piece hanger
Bias spring
Spring retainer
A
A
Slot for installing poppet; poppet inserted on edge and rotated to desired position
View A-A
Retaining sleeve
Holders inserted from both sides through holes in hanger after compressing spring
Poppet being inserted
Fixture for holding spring while installing poppet

(b) Successful construction.

Figure 19. - Inlet valve subassembly construction. Concluded.

2.2.5 Control Elements

The relief valve may include a variety of control elements, the number and type depending on the complexity required to meet the specified performance. All relief valves have a spring reference load that is adjustable to the relieving pressure range; thermal compensation for the spring may be provided. Orifices are used to restrict the flow of fluids used in pilot circuits and venting control. The problems with the reference load spring, thermal compensator, and orifices are similar to those discussed in section 2.1.4. Some of the tank relief valves are large, and operation of the unit can be affected by duct size and length and by interaction of the control elements. A computerized mathematical model of the system may be used to evaluate transient conditions and indicate possible solutions (ref. 54).

2.2.6 Operational Problems

Relief valves are subject to the kinds of operational problems discussed for regulators in section 2.1.5; in general the solutions given for regulators apply equally well to relief valves. Additional problems with relief valves involve moisture in the actuator, pilot valve pressure sensing, and operational stability.

2.2.6.1 MOISTURE IN THE ACTUATOR

In many tank relief valves, an actuator controlled by a separate pneumatic pressure source opens the main inlet valve to vent the tank. A hydrogen-tank relief valve had low leakage under normal conditions but leaked excessively when exposed to cryogenic conditions; the failure was random. After careful investigation, the failures were attributed to moisture in the clearance between the actuator and its deactuated stop. In initial chilldown of the unit, the moisture froze and held the actuator pushrod in contact with the inlet valve. With continued chilldown, the differential contraction of the parts lifted the inlet poppet from the seat and caused the excessive leakage. Procedures were established to ensure that the helium control fluid for the override actuator was dry. Also, new units were vacuum-dried at elevated temperatures for minimum periods of time and stored in dry sealed polyethylene bags.

2.2.6.2 PILOT VALVE PRESSURE SENSING

The relief valve pressure sensor is a pressure-actuated device that creates a force opposing that of the reference spring. In the direct-operating valve, the inlet poppet senses the pressure, and the reference spring opposes its motion. In the inverted-operation valve, a piston, bellows, or diaphragm senses the pressure, and the reference spring opposes its

motion; this piston, bellows, or diaphragm is the actuator of the inlet valve. The pilot valve sensor usually is the pilot poppet actuator, and the reference spring opposes its motion.

2.2.6.2.1 Sensor Location

In a piloted, large-tank relief valve where the pressure is sensed within the valve, the location of the source or pressure to the pilot can affect the operation and response of the valve. Where the pilot valve exhausts into the outlet, its exit port location is important. For example, in a vent-and-relief valve for a liquid-oxygen tank, the pilot sensed pressure in a low-flow area in the body; it exhausted into the outlet where dynamic ram pressures during relieving created back pressure on the pressure-unbalanced pilot. The response of the unit was slow, and allowable relieving-pressure limits were exceeded. The pressure measured at the pilot sensing diaphragm differed from tank pressure. The pilot exhaust was changed to eliminate back pressure on the pilot, and valve operation and response were greatly improved. The optimum location for pilot valve inlet pressure sensing under flowing conditions was determined by scanning the entire inlet port area with a total-pressure pickup. Where the pickup sensed the total pressure, the pressure measured at the pilot sensing diaphragm very closely reflected static tank pressure; the pilot valve sensor was relocated at this point. With these changes, the operation of the relief valve met the specified performance (ref. 55).

2.2.6.2.2 Sensor Material

In many relief valve applications, the external joints are sealed by welding to ensure near-zero external leakage. Metal diaphragms are used because they may be welded and are not damaged by exposure to temperatures resulting from valve closure welds. The metal selected must be compatible with the fluid with which it is in contact. In one instance, a hydrogen-tank vent-and-relief valve used a thin, annealed 17-7 PH stainless steel sheet diaphragm that apparently operated successfully during extensive static engine testing, when the gaseous hydrogen sensed by the valve was supplied by the facility. When a new series of static engine tests was conducted, and the tank was refilled from the engine liquid-hydrogen supply, the valve diaphragm ruptured. Several such failures led to an investigation of the effects of the application and environment on the diaphragm. The test results indicated that exposure of thin sheet 17-7 PH material to hydrogen under simulated-use conditions gradually increased the hydrogen content until failure occurred as a result of hydrogen embrittlement (ref. 56). Test exposures of annealed Inconel 718 diaphragms to similar conditions indicated that, even though the hydrogen content also increased, the Inconel 718 is not embrittled under the same conditions as the 17-7 PH stainless steel. The diaphragms fabricated from Inconel 718 operated successfully in subsequent engine tests and during Saturn S-IVB orbital coast.

2.2.6.3 STABILITY

A control system must be dynamically stable throughout its entire range of operation. A system can become dynamically unstable when the pressure and flow fluctuate in a sustained oscillation. When two pilot-operated devices are in series, their interactions can be 180° out of phase and thereby produce dynamic instability. For example, a pneumatic regulator and relief valve were developed for use in a large-engine operational control system; the regulator was a normally closed unit. The system pressurization was initiated by energizing a pilot solenoid valve that supplied pressure to the inlet valve actuator opening volume to open the inlet valve rapidly. A resulting pressure ramp in the regulated pressure would rupture the regulator controller diaphragm unless the relief valve prevented the overpressurization. The regulator and relief valve were combined into one housing to save weight, plumbing, and possible leakage paths. When the combined unit was tested, a sustained unstable interaction of the units prevented the buildup of system pressure.

Figure 20 shows a cutaway of the regulator–relief valve assembly and indicates the flow

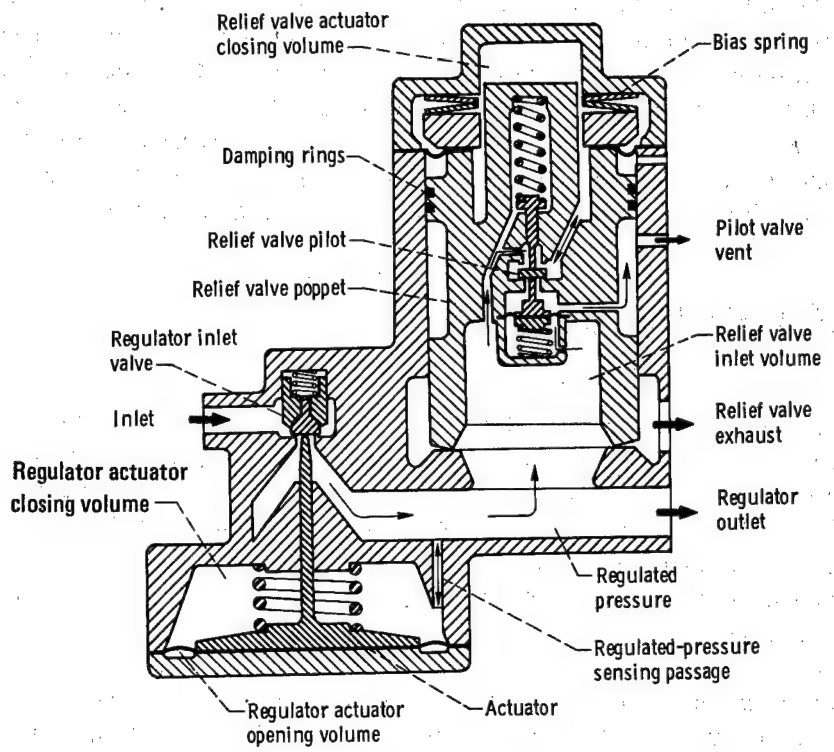


Figure 20. - Schematic of a regulator-relief valve assembly.

passages within the unit. An assembly was modified to add pressure taps in the critical control volumes. A series of tests was conducted, pressures were recorded, and valve

behavior was observed. The data were analyzed by analog and digital computer programs. The analyses indicated three possible solutions:

(1) Reduce the actuator closing volume and increase the regulated-pressure sensing passage (fig. 20) of the regulator to decrease the magnitude of the regulated-pressure surge during the starting transient.

(2) Enlarge the relief valve inlet volume to tune the circuit and eliminate the out-of-phase pressure surge in the actuator volumes.

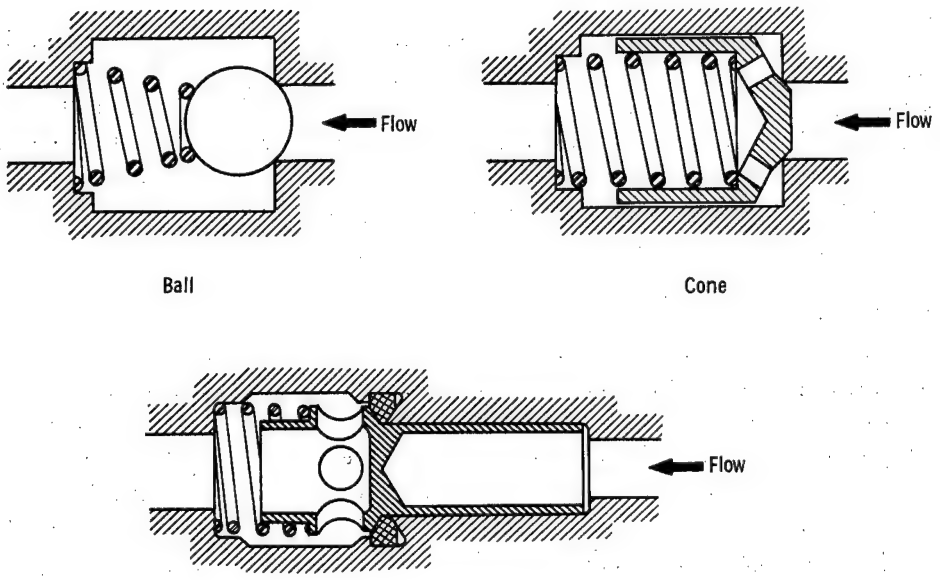
(3) Increase the rate of pressure rise in the relief valve actuator volume by increasing the pilot valve seat diameter; delay the opening of the relief valve main poppet by increasing the bias-to-close spring force; add damping rings to the relief valve poppet and reduce the stroke of the relief valve poppet.

The changes outlined in the third solution were implemented. The effect of varying the damping-ring friction was studied to determine the minimum amount that would give the desired stability. The regulator operation was stable. During the start transient, the relief valve opened several times to prevent the overpressurization of the regulator controller diaphragm and then closed. At the time of the implemented changes to the unit, a large accumulator was added to the system pressurized by the regulator. Additional computer runs, as well as system tests, indicated that the damping rings could be eliminated from the main poppet. The damping rings were retained, however, as an added stabilizing feature.

2.3 CHECK VALVES

The primary function of a check valve is to allow flow in one direction only. Check valves are divided into two basic types: poppet and flapper; both types are shown in figure 21. In poppet-type units, the moving parts travel axially; springs are coil-type compression; and the poppet may be a ball or a guided poppet with a spherical, conical, or flat nose to contact the seat for sealing. In flapper-type units, moving members rotate; springs are torsion type; and the flapper may be a single unit that pivots from the side or a split unit that pivots across the center of the flow path.

In the checked-flow direction, a good check valve will have low leakage over the entire pressure range of operation. Check valves are pressure-unbalanced and, under high-pressure conditions, high unit seat loadings are possible. Most leakage problems occur at low pressures where the primary seating force is exerted by the spring. The spring force is small to allow low cracking pressure; therefore, low leakage rates are achieved by the design of the seat. Recommended practices for design of poppets and seats are given in reference 26.

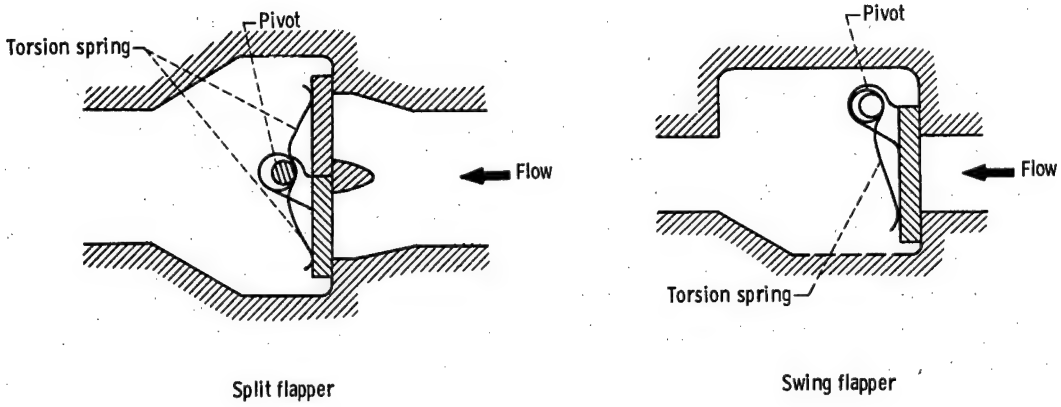


Ball

Cone

Floating ring

(a) Poppet.



Split flapper

Swing flapper

(b) Flapper.

Figure 21. - Types of check valves.

2.3.1 Poppet

Poppet check valves (fig. 21(a)) are used in applications requiring low leakage in the checked-flow direction and reasonably low pressure drop in the free-flow direction. The cracking pressure is low (3 to 15 psid [21 to 103 kN/m^2]) and the full-flow pressure is as close to the cracking pressure as possible. The cracking pressure of a poppet-type check valve can be increased by increasing the return-spring installed load. Under these conditions, the unit functions as a relief valve or a back-pressure regulator.

2.3.1.1 BALL

Ball check valves are simple devices that are relatively inexpensive to manufacture, as the seat is the only costly detail. The ball check valve is used in low-flow applications where the stroke is short and the ball will self-guide into the seat. This type of valve generally is not used for large flows because the long stroke allows misalignment and the ball, when closing, may impact the side of the seat; the seat and ball will be damaged by repeated impact and will leak excessively. The addition of a moving part to guide the ball into the seat complicates the valve and makes it equivalent to other guided-poppet-type valves. Ball-type check valves are prone to chatter.

2.3.1.2 GUIDED POPPET

The purpose in guiding a poppet is to align the poppet so that contact with the seat will be continuous and even. With flat-nose poppets, the clearance with the bore and the normality of the flat nose must be controlled carefully for a continuous and even contact with the seat. The normality problem is eased by using the floating seal ring shown in figure 21(a). The cone of the cone-type poppet must be controlled carefully to obtain a seat contact line that is round and in the plane of the seat. For adequate guiding, the clearance must be small. Although the surfaces of poppet and bore may be lightly loaded, they must be hard enough to resist wear and fretting. If flow rate is high over long periods, the fluid flow is directed around the return spring and not through it. In an engine-conditioning check valve on the J-2 engine, flow directed through the return spring caused spring flutter that resulted in excessive valve wear.

The restrictor check valve (fig. 22) is a specialized use of the checking function. An orifice is provided in the poppet to permit a controlled flow in the check direction. Leakage past the poppet seat is critical if the reverse flow must be controlled within narrow limits.

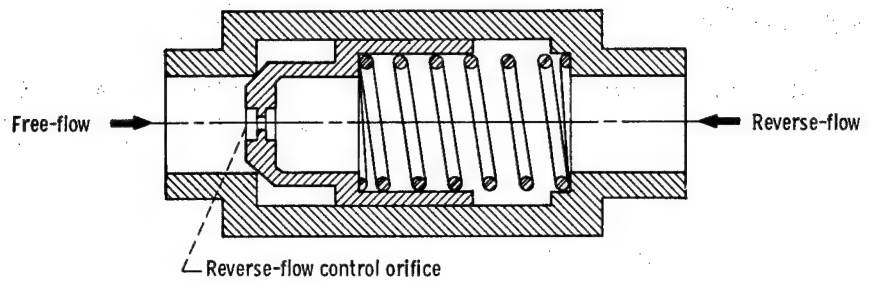


Figure 22. - Restrictor check valve.

2.3.2 Flapper

Flapper check valves (fig. 21(b)) are used in systems requiring low pressure drop with large flows. These units tend to be smaller and lighter than the poppet-type check valves. However, in high-pressure systems, the flappers may become excessively heavy.

Full-flow area is achieved by the swing-flapper check valve when the flapper swings or pivots out of the flow passage. The low installed load and spring rate of the torsion spring permit a low cracking pressure and pressure drop. With sufficient clearance in the pivot, the flat flapper and seat have good sealing characteristics at elevated pressures.

One cause of failure of flapper check valves is chatter of the unit, which occurs when the flow is low. Extensive damage to the flapper and seat also can result from large impact forces if angular travel is excessive; and failure of the torsion spring can result in a loss of the check function. The flapper is subject to "hang up" because the spring forces are low.

Split-flapper check valves are used when the envelope dimension is critical and low pressure drop is required for large flows. When leakage is not critical, each flapper seals on half of a common seat, and leakage between the flappers is controlled by the clearance. When leakage is critical, a special seat for sealing each flapper individually is provided by a land across the flow passage, in line with the pivot of the flappers (ref. 57 and fig. 21(b)). The upstream side of this land is streamlined for minimum pressure drop. Split-flapper valves also chatter at low flow and in general are subject to the same kinds of potential failure as the swing type.

2.3.3 Operational Problems

The application of the check valve is a source of possible problems. Incorrect installation, chatter, and slow closing can adversely affect the system in which the valve operates. Contamination and moisture also have caused failures.

2.3.3.1 INSTALLATION

Check valves frequently are used to provide single-direction flow in systems like that in figure 23. During engine start, the start-tank check valve permits flow to the turbine-drive

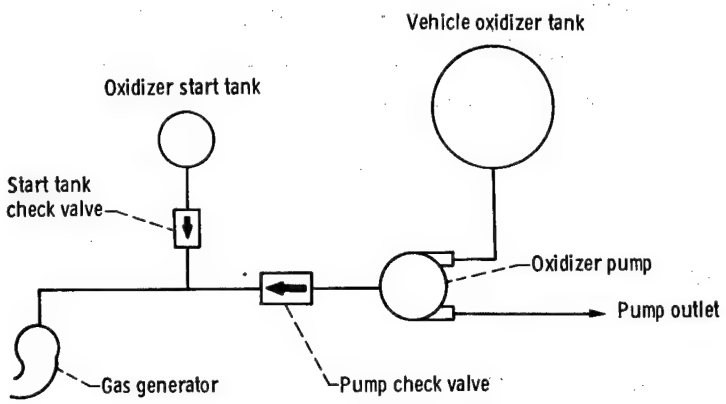


Figure 23. - Schematic of engine start system.

gas generator, while during mainstage the pump check valve permits flow from the main tanks to the gas generator.

A basic problem with check valves is the possibility of system failure resulting from wrong-way installation. As an example, prior to one vehicle launch, the pump check valve was removed for an inspection check and reinstalled backward. When the launch was attempted, the gas generator fired until the start tank was exhausted; then the engine shut down. The mission was aborted and the launch facility was damaged.

To prevent this type of failure, check valves are permanently marked with an arrow showing the direction of flow; however, in order to install the unit correctly, the direction of flow in the system must be known. To ensure a truly "goof-proof" installation, the ends of the check valve are made different in size or construction.

2.3.3.2 CHATTER

Poppet check valves will chatter when opened by a low flow that does not fully stroke and hold the poppet in the full-open position; low-flow purges may cause a poppet to chatter for many hours. Series redundant check valves accentuate the tendency to chatter because of valve interaction. The repeated actuation in a chattering valve can exceed the normal operating life and so damage the seat and sliding surfaces that the unit will fail. Wear and fretting of the sliding surfaces by excessive actuation have generated contamination that

caused failure of other units downstream from the check valve. The tendency to chatter has been lessened or eliminated in some designs by incorporating a restriction in the flow passages around the poppet; the resulting pressure differential applied to a sufficiently large opening area holds the poppet off the seat during low flows (ref. 58). If chatter cannot be eliminated entirely, it is possible to increase the check valve tolerance to withstand chatter damage by providing very hard bearing surfaces; hard anodize, properly controlled and finished, is an effective wear-resistant surface.

Chatter at low flow and under other operating conditions is a common problem with flapper-type check valves. Two examples illustrate design solutions for chattering flappers:

> In one application, the flapper pivoted into a recess out of the flow passage; however, the dynamic forces on the flapper produced by the flowing fluid were insufficient to hold the flapper in the recess. The rattling flapper failed at the pivot and permitted high-pressure fluid to bypass into the return system. A stop was added to hold the flapper in the fluid stream; the increase in pressure drop was negligible, and the flapper did not chatter.
>
> A split flapper was exposed to flowing high-temperature combustion products; because of the high temperature, the torsion spring relaxed and would not close the flappers. Leakage was high until the dynamic flow forces in the check direction closed the valve; this leakage exposed components to a high temperature and contamination for which they were not designed. The spring was redesigned to improve its yield characteristics and maintain the desired load at high temperature; in addition, a shield was added to protect the spring from flowing hot gas.

2.3.3.3 POPPET CLOSING TIME

When propellant tanks are pressurized from a single source of gas, check valves are used to prevent backflow of the propellants into the common pneumatic pressurizing system. Under normal operating conditions when tanks are refilled from a turbopump, the pressure of the fuel tank exceeds the pressure of the oxidizer tank. It is important therefore that the check valves close as soon as the tanks are pressurized and refill starts. Delay in closing will result in crossflow of propellants and the potentially disastrous introduction of fuel in the oxidizer tank. For example, in the Atlas vehicle, when the poppet of the check valve in the fuel system stroked to the open position, a seal was effected between the poppet and stop, and a low-pressure condition existed in the reference-spring cavity of the check valve. When closing flow started, the valve stayed open for a period of 5 to 20 seconds until leakage of fluid into the spring cavity equalized the pressures and the poppet closed. When engine start was attempted, an explosion occurred in the oxidizer tank and connecting line to the gas generator and caused extensive damage. Addition of a hole in the web isolating the reference-spring cavity from the downstream area eliminated the slow closing of the poppet.

2.3.3.4 CONTAMINATION

Poppet check valves with close-clearance guides are subject to failure if the flowing fluids are contaminated. Solid-particle contaminants cause friction in the sliding surfaces, and the poppet may stick in the open position because the small actuation force of the spring is not sufficient to release the poppet. Contaminant particles may damage the poppet and seat, or hold the poppet off the seat and cause excessive leakage. Contamination in the system can be kept from the valve by the addition of a filter upstream of the check valve inlet; the capacity of the filter is such that there is no excessive pressure drop (ref. 47).

2.3.3.5 VENTING

In pneumatic systems, the gases used to pressurize valve actuators are exhausted to atmosphere as the valve opens and closes. Cavities of valves that contain actuator linkages have vent ports to exhaust possible actuator piston seal leakage. These vent ports are capped with a check valve (vent port check valve) to prevent moisture and contaminants from entering the critical areas of the valves. Small amounts of frozen moisture or other solids in the close clearances of actuation mechanisms located in these cavities have prevented main propellant valves from closing. Failure of these valves to close on shutdown signal resulted in extensive engine damage.

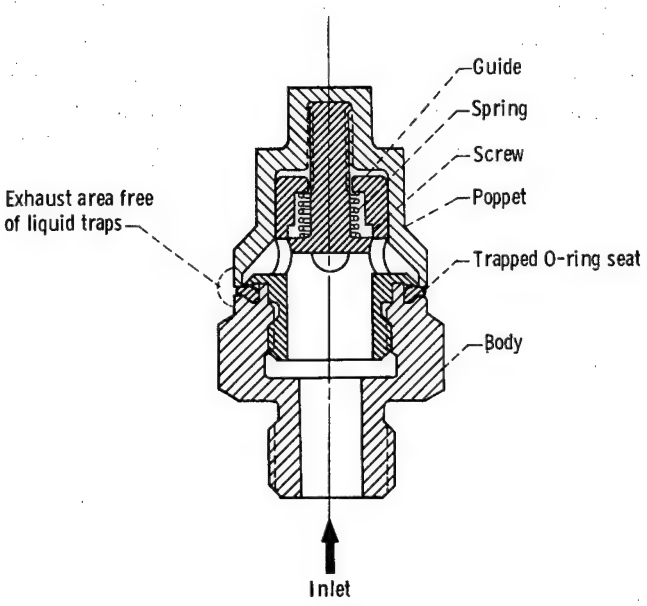
The design of the vent port check valve is critical. The checking poppet is an external member that prevents contamination of the moving parts of the check valve. In cryogenic applications, ice builds up on the external surface of the poppet where it contacts the valve body. The motion of the poppet is designed to create tension stresses in the ice. If the ice must fail in shear to permit the poppet to open, the cracking pressure will be undesirably high.

The construction of the unit is rugged to prevent damage to sealing areas. Vent port check valves are subject to rapid opening with high impact loads on the open-position stop. The open-position stop is designed for high impact loads, and threaded fasteners are provided with positive locking devices.

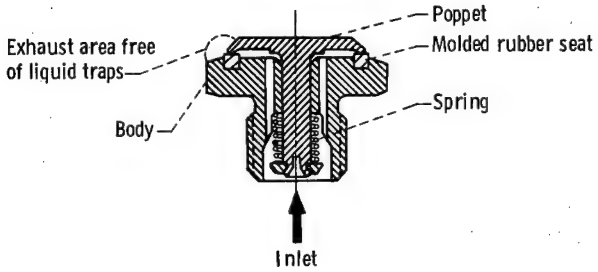
Successful designs for high- and low-capacity vent port check valves are shown in figures 24(a) and 24(b). The molded Teflon plug type shown in figure 24(c) was used in early space vehicles. The sealing surface of the boss and plug was damaged easily, and the units frequently leaked.

2.4 BURST DISKS

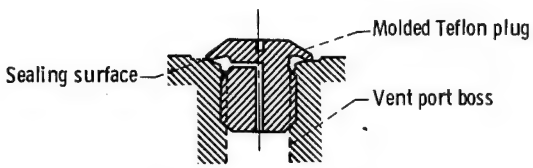
Burst disks, also known as rupture, frangible, and safety disks (or diaphragms), have been used extensively as safety devices in the chemical and petroleum industries. These disks



(a) High-capacity assembly (acceptable).



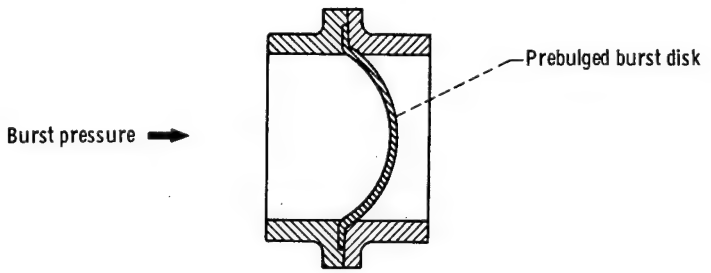
(b) Low-capacity assembly (acceptable).



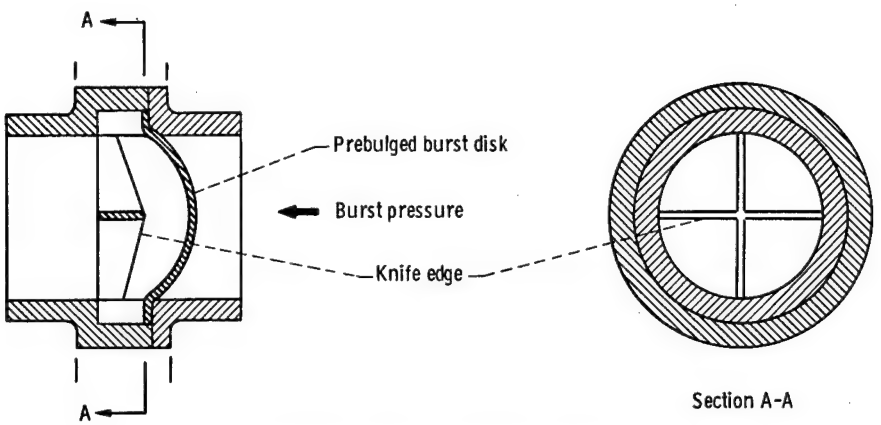
(c) Low-capacity plug (unacceptable).

Figure 24. - Types of vent port check valve.

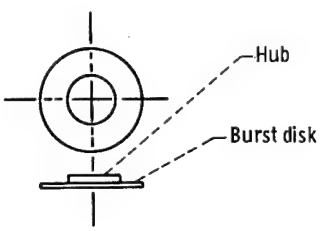
usually are the prebulged type (fig. 25(a)) and fail in tension without the use of local weakening such as coined or machined grooves to initiate and control the bursting. A considerable amount of theoretical and experimental work (refs. 59 through 61) has been done on prebulged burst disks. Even though a theoretical analysis is possible, the results are only approximate, and it is necessary to arrive at the final design by actual test; the manufacture then is controlled by sampling. Variations on the prebulged type of burst disk include disks that reverse form and buckle onto a knife edge (fig. 25(b)), thus bursting at a lower pressure.



(a) Prebulged burst disk.



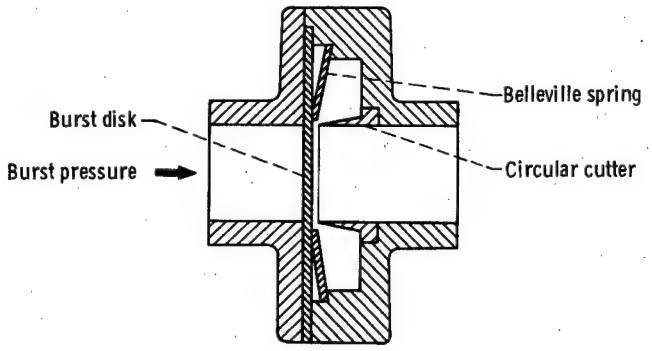
(b) Reverse-action knife-blade burst disk.



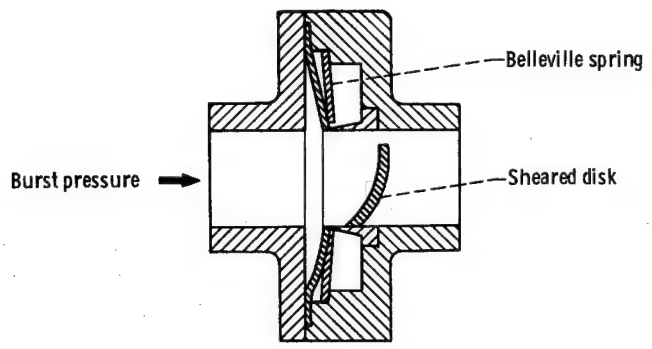
(c) Integral-hub shear burst disk.

Figure 25. - Types of burst disks.

Shear-type burst disks also have been utilized in the chemical industry (ref. 62). Shear disks however, generally are considered to be less precise than the prebulged type; they do not fail in pure shear but have some bending stress present because the clearance between the holder and central hub stiffens the center of the disk against bending. Also, the shear disk is more costly because it must be turned out of thicker material so that the thick hub can be left in the center (fig. 25(c)). The inertia of the disk is high. Some of the objections to the shear-type burst disk are overcome in the design shown in figure 26 (ref. 63). This design pressure-loads the disk on a shearing cutter by the use of a Belleville spring that is



(a) Disk prior to rupture.



(b) Disk after rupture.

Figure 26. - Improved shear burst disk (ref. 63).

compressed into the negative-spring-rate region of the spring stroke. The Belleville spring supports a portion of the pressure load on the burst disk up to the desired burst pressure, as shown in figure 26(a). When the burst pressure is exceeded, the spring washer starts to stroke, its supporting force decreases rapidly, and the pressure force causes the disk to be sheared by the sharp circular cutter (fig 26(b)). A catch screen may be required to keep the cut disk from traveling downstream.

The burst pressure can be preset by replacing the cutter with a flat plug and adjusting the Belleville spring washer until the desired burst pressure is indicated by the motion of the burst disk. The burst pressure can be set within a narrow tolerance. It is not necessary to burst a controlled sample, because the Belleville spring load compensates for variables in disk materials. Problems can arise, however. For example, in one application of this design, the disk was corroded and weakened by residual cleaning solvent trapped in the unit; a change of disk material eliminated this difficulty.

The development of fluid systems for space vehicles resulted in the application of burst disks in many areas, including rocket engine hypergolic-start cartridges, repeat-start turbine spinner assemblies, pump-seal drain lines, and some instrumentation. The emphasis also changed from using burst disks as safety devices to using them in various functional applications (e.g., sealing a cavity against downstream pressure or temperature, or containing a liquid until a rise in pressure to a desired level ruptures the burst disk). The use in space systems also resulted in increasing the severity of the design requirements by introducing the need for compactness and lightness, extending the temperature range to cryogenic temperatures, and minimizing the allowable tolerance range. The result has been the development of the improved shear disk shown in figure 26 and the flat, coined-groove, welded-installation disk in figure 27. (The chamfered notch and locally enlarged opening shown in fig. 27 are discussed later.)

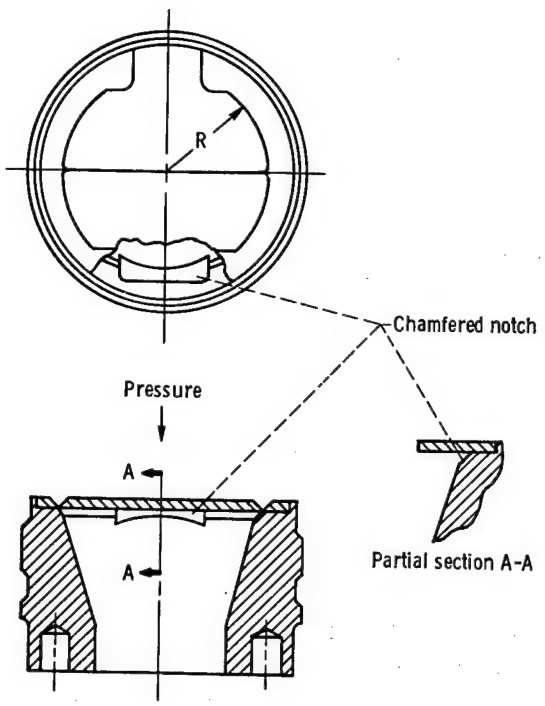


Figure 27. - Flat, coined-groove, welded-installation burst disk.

2.4.1 Burst-Pressure Capability

The burst pressure is the primary element in the design of a burst disk. The burst pressure must be high enough that the burst disk will remain intact under all conditions of storage, handling, and use until the system pressure rises to the point at which it is desired to have the disk burst. At this point, the disk must burst within the allowable tolerance. The requirements of the balance of the system usually define the burst pressure, which may range from 1 to 10,000 psid (6.895 to 68,950 kN/m^2).

Some burst disks are designed to burst at as low a pressure as possible so that they effect a function at a low system-pressure level. This would be the case in a cartridge containing an ignition aid for a rocket thrust chamber where it would be desirable to burst the cartridge disks, release the ignition aid, and establish ignition as soon as possible in order to minimize starting shocks and unburned propellant. Ignition pops and rough-combustion starts can cause extensive thrust-chamber damage. The burst pressure must not be so low as to result in premature failure due to vapor pressure, thermal expansion of the propellant, or handling loads.

On the other hand, when a burst disk is used to ensure rapid, repeatable ignition of a pyrotechnic device, it is desirable to set the pressure as high as possible so long as the burning rate of the charge does not become excessive and the pressure level remains below the allowable operating pressure of the container. The rate of pressure rise can affect the burst pressure; when burst disks are tested on a percentage basis, the rate of rise of applied pressure is made the same as in the actual application.

In cases where shock is important because of its effect on the rest of the system, the burst pressure may have to be low so that initial flows occur at low levels of pressure and hence at low velocity.

2.4.2 Reverse-Pressure Capability

The reverse-pressure capability of a burst disk is as important in some applications as the burst-pressure capability. In the case of the prebulged disk, the disk resistance to reverse pressure is very much lower than its resistance to the burst pressure. This characteristic is used to lower the burst pressure for a given disk by reversing the pressure and buckling the disk onto knife blades as shown in figure 25(b). The flat coined-groove disk has a much greater capability to resist buckling by reverse pressure, because it is a thick flat member with local thinning at the grooves and is not a uniformly thin dome.

The ability to resist reverse pressure is important in disks that protect vacuum jackets around a high-pressure system against leakage from the system. Also, in the case of solid propellant charges used for repeat starting of rocket engines, the unused charges must be

protected from the hot high-pressure gases by burst disks that will resist reverse pressure but will burst when the solid charge is ignited. These dual requirements sometimes are difficult to meet. One solution is to support the burst disk against reverse pressure with a structure that is ejected or does not seriously impede flow when the disk is burst.

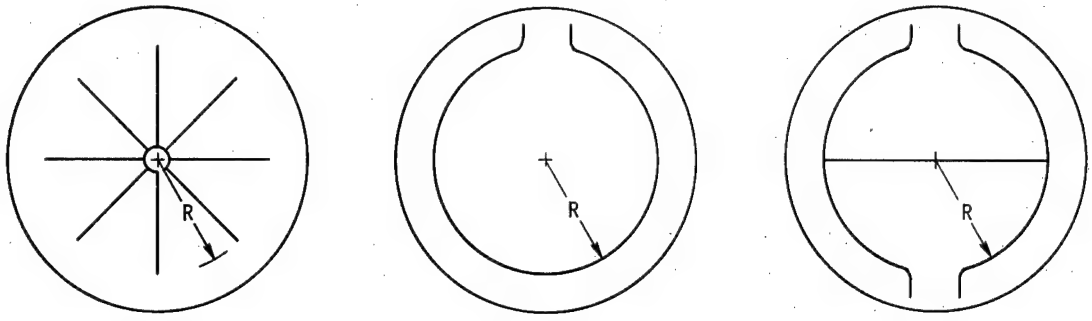
2.4.3 Size and Thickness

The diameter of the burst disk normally is large enough that no significant pressure drop is imposed when the disk is open. This may require that the downstream line into which the disk opens be enlarged locally as shown in figure 27. There is a requirement for increased disk diameter that often is overlooked in an effort to minimize the envelope; namely, that in the coined-groove disk the force required to rupture the disk is a function of the area lying inside the coined groove, while the disk resistance to rupture is a function of the length of the coined groove plus some bending at the hinge. Thus, for a given thickness of material and depth of coined groove, the larger the disk the lower the burst pressure. Similarly, for a fixed burst pressure, if the disk can be enlarged, the thickness of the disk itself and the thickness of material remaining after coining can be made greater and less sensitive to a given tolerance variation. The required tolerance on the burst pressure decreases as the diameter increases, while the range of material thickness that can be used increases; closer tolerance can be held, but only with extra care, scrappage, and cost. The same principles hold for the prebulged type of burst disk.

Best results in sizing the disk are obtained when burst disk is included in the original system design and sufficient space provided to yield the best burst disk characteristics. The addition of a burst disk late in a program may limit the size of unit that can be used. The material thickness is determined largely by the disk diameter and (desired) burst pressure and thus cannot be specified as an independent parameter. An increase in thickness results in greater margin for corrosion or accidental damage, in easier fabrication and handling, and in relaxed tolerances. For coined groove disks in common use, the thickness remaining under the groove ranges from 0.002 to 0.125 in. (0.05 to 3.18 mm).

2.4.4 Coined-Groove Pattern

In the coined-groove burst disk, the material is weakened locally by grooves that are stamped into the metal. Three groove patterns are in common use: radial, single hinge, and double hinge (fig. 28). The radial type (fig. 28(a)) has radial grooves converging in the center at a small circular groove that is interrupted at one sector for retention of the center after burst. This type of ruptured disk produces less flow disturbance because the flow is centered, but has a higher pressure drop because the disk does not open fully. The higher pressure drop is suited for pyrotechnic applications because the initial, restricted opening aids in maintaining a back pressure that ensures quicker, more consistent, reliable ignition.



(a) Radial. (b) Single hinge. (c) Double hinge.

Figure 28. - Coined-groove patterns (ref. 64).

The subsequent burning off of the petals opens the disk fully; the effect of this debris on the downstream system must be considered in the design.

The single hinge (fig. 28(b)) is better able to resist reverse pressure but concentrates the flow disturbance on the side of the open petal. The double hinge (fig. 28(c)) has the lowest tolerance on burst pressure and divides the flow disturbance.

2.4.4.1 PETAL RETENTION

The function of the hinges in the coined-groove pattern is to retain the petals after bursting. Petals that have broken loose and, at high velocities, impacted downstream components such as filters have caused considerable damage. Failure of valves can be caused by petals wedging in moving parts. The hinge must be wide enough to retain the petal at burst and during subsequent flow through the disk; if shock waves can occur, the possibility of flutter and fatigue of the hinge must be considered. As the width of the hinge increases, however, the curvature of the line into which the petal breaks puts the edges of the hinge into increasing tension and causes cracks; then it becomes necessary to add a chamfered notch under the hinge (fig. 27) to provide a smooth transition surface and thereby reduce bending stress on the hinge. Petal flutter also can fatigue the hinge when a line discharges into an enlarged cavity containing the opened disk, and high-velocity gas does not impinge on the petals. A conical downstream cavity (fig. 27) that closes down in the direction of flow has been used to hold down the petals. Better petal retention is obtained with the coined groove facing upstream, since this position puts the torn edge of the hinge in compression.

2.4.5 Material

Burst disks have been made from aluminum alloys including 1100-0, 6061-0, and 5052-0, mild steel, stainless steel (usually the 300 series), nickel, Monel, Inconel, copper, silver, gold,

and platinum. Aluminum is desirable because its relatively low physical properties result in greater thickness of material and increased thickness remaining under the coined groove, and therefore less sensitivity to a given tolerance; forming and machining characteristics also are good. However, the material properties vary widely with temperature, and the upper temperature limit is only about 250°F (394 K).

Mild steel also is good if suitably protected from corrosion; the material cost is low, and formability is good. Stainless steel is difficult to tool, as is Inconel; the latter is best for service over a wide range of temperature. The noble metals resist corrosion and have consistent material properties.

The method of attachment enters into the choice of material in that mechanical attachment allows a wide choice, whereas welding limits the choice. The material selection must also consider the fluid in contact with the burst disk, because corrosion of the burst disk by the fluid or decomposition of the contained medium by the disk must be avoided. Corrosion cannot be tolerated, because the thin sections of the burst disk have no margin for loss of material and the burst pressure would be reduced (ref. 65). Decomposition of the contained medium could result in a pressure buildup that would cause premature rupture of the burst disk.

2.4.5.1 TEMPERATURE EFFECTS ON PROPERTIES

Most mechanical properties of a metal usually are reduced by an increase in temperature; the reverse occurs when temperature is decreased. Because the properties affect the burst pressure, it becomes necessary to determine the operating temperature accurately and to adjust the ambient burst pressure to reflect the change in properties with temperature (ref. 66).

Inconel is a desirable material for service at high temperatures because the reduction of burst pressure with increasing temperature is less than that for materials such as aluminum or stainless steel (fig. 29). Although data for burst disks at low temperatures are not available, the variation in burst pressure should parallel the variation in tensile strength shown in figure 30 (ref. 66). The strengths of aluminum, stainless steel, and Inconel decrease by factors of 6.8, 2.6, and 1.5, respectively, between -300°F (89K) and 500°F (533K). If aluminum is used because burst pressure is low or diameter is small, then a lower temperature limit and greater variation in the burst pressure must be accepted. At low temperatures, regions of brittleness may exist with some materials (e.g., low-carbon steel). At high temperatures, the creep of the material becomes important and may dictate lowering the operating pressure to widen the margin between operating and rupture pressures.

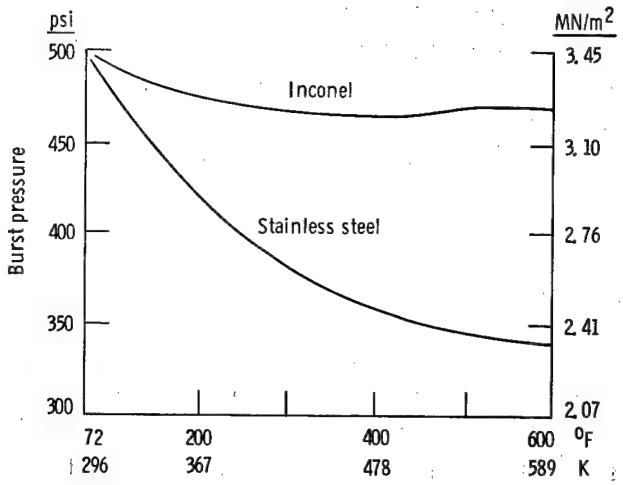


Figure 29. - Variation of burst pressure with temperature.

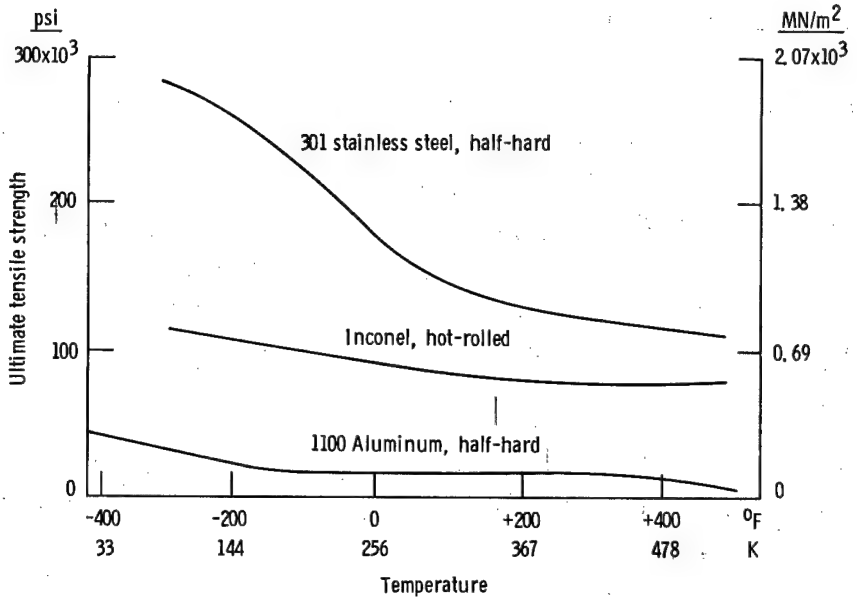


Figure 30. - Variation of material ultimate tensile strength with temperature (ref. 66).

2.4.5.2 FATIGUE STRENGTH

The linear cumulative damage rule (ref. 67) is used to estimate the life fraction consumed when a material is cycled through a known stress or strain range. This cumulative effect is of concern when burst disks are used for purposes of relieving excess pressure but is not applicable when only one operation and not repeated cyclic service is required. In any case, any application and removal of pressure or fluctuation in pressure is evaluated for possible fatigue effects. The usual precaution is to raise the burst pressure with respect to the operating pressure.

2.4.6 Installation

The installation of prebulged burst disks is a simple matter if the manufacturer's instructions are followed and if good practices are used, because such disks usually are attached mechanically. To ensure correlation of burst pressure with test results, the test fixtures must duplicate the clamping means actually used on the vehicle. When the disk is attached by welding, as is the case with most flat coined-groove disks, the thermal stresses set up by welding can prestress the coined section of the burst disk and lower the burst pressure. In addition, the thin burst disk tends to become overheated before the heavier body is heated, and this difference intensifies the thermal stresses. One successful method of welding the burst disk is shown in figure 31.

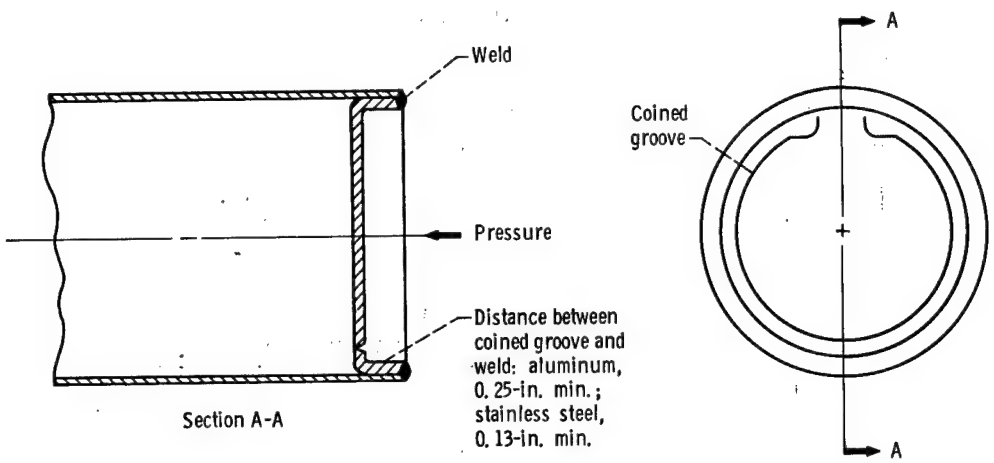


Figure 31. - Cup-type, welded-installation, coined-groove burst disk (ref. 64).

Because burst disks are somewhat specialized elements with the unusual requirements that they bend and break in order to function, their design is not as amenable to analysis as one where the part must not break and can therefore be provided with a margin of safety. A burst disk with a margin of safety would not be useful. Consequently, many more variables come into play – the effect of work hardening during forming, stresses set up by installation, and temperature effects on material properties – all in a dynamic situation where the form and dimensions of the disk are changing as it functions. Very early coordination and cooperation with the supplier are essential to the efficient, timely design of burst disks. Burst disks arbitrarily designed into existing spaces and parts without allowing the supplier to influence the design have a tendency to be costly and troublesome; for example, with a welded-in coined-groove disk, a lack of adequate space can cause the distance between the groove and weld to be too small, or it can make the welding difficult because of angle or position.

Burst disks may be installed in locations where they are subject to denting or other mechanical damage due to dropped or flying objects. This installation may be the final one or may occur during assembly or servicing. The disk can be damaged sufficiently to affect the burst pressure, especially if the disk is thin. An outlet line with a "dogleg" that points down prevents most kinds of mechanical damage.

2.5 EXPLOSIVE VALVES

Explosive valves, also known as squib valves, have been used primarily in space applications. Their generally nonreusable nature makes these valves undesirable for commercial applications, whereas their zero leakage, small size, light weight, rapid response, and self-contained actuation requiring only a small pulse of electrical energy make them most useful for one-shot space requirements. In deep-space vehicles, a system must be stored or isolated for long periods of time without leakage. With normally closed and open valves in series, it is possible to activate and deactivate a guidance system for a short period of time by energizing first the normally closed valve and then the normally open valve. A series of sequences can be performed by combinations of valves within a small envelope and weight. A variety of configurations such as normally open, normally closed, three-way, four-way, and combinations of two normally closed valves in a single body or normally open and closed valves in a single body are available. Examples of several commonly used configurations are presented in figures 32 through 35.

Figure 32 shows a normally closed valve that opens when the explosive charge actuates a pin (ram) that shears the end from the inlet fitting and retains it in a recess in the body. This unit is made reusable by replacing the explosive charge and inlet fitting; reusable units must be cleaned immediately after firing. Another normally closed valve is shown in figure 33 as it appears after firing. A diaphragm is sheared from the body by the ram and is clamped by the ram to retain it within the body; this unit is not reusable.

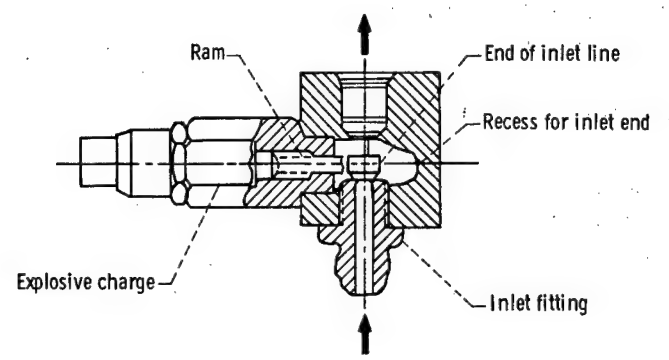


Figure 32. - Normally closed reusable valve (before firing).

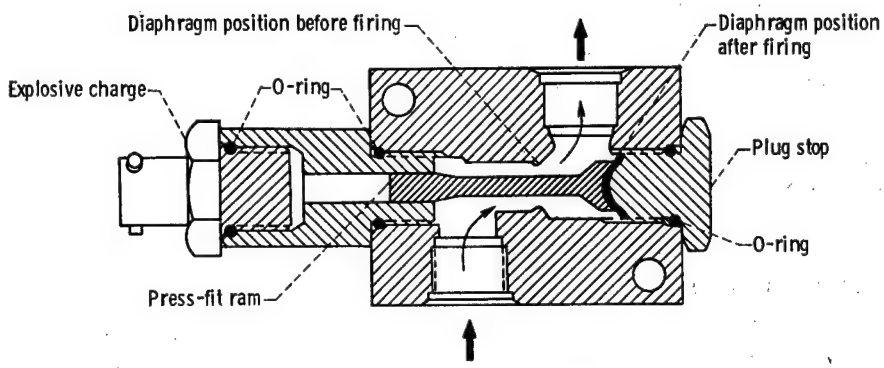


Figure 33. - Normally closed valve (after firing).

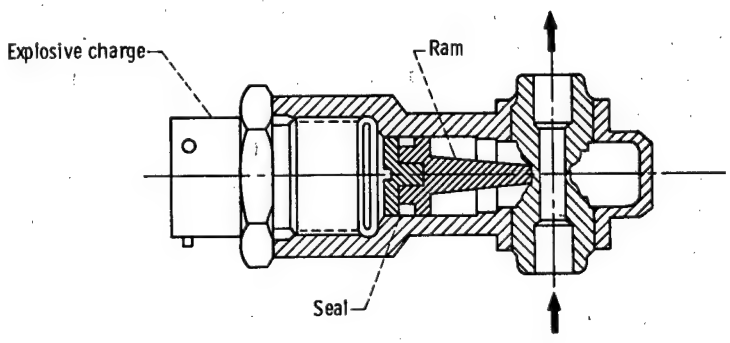
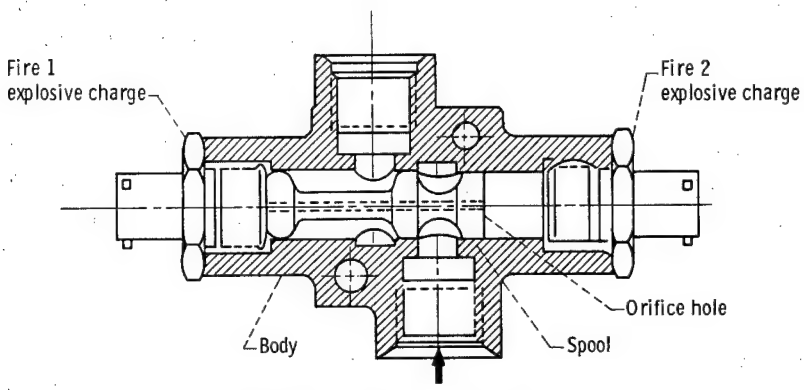
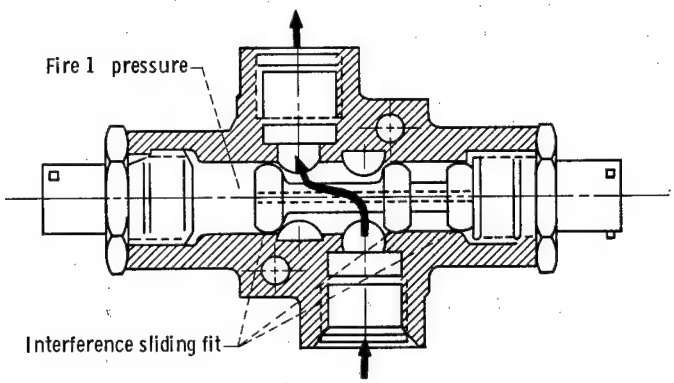


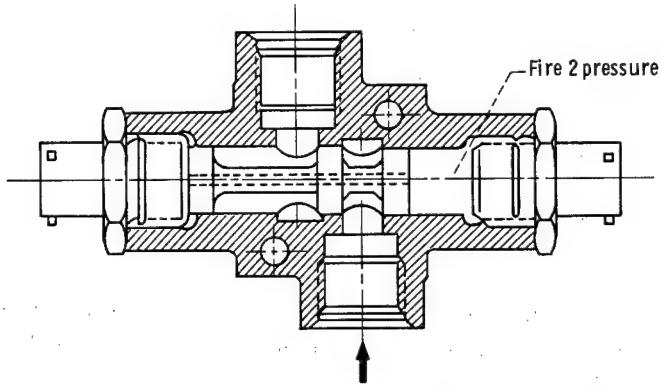
Figure 34. - Normally open valve (before firing).



(a) Normally closed valve before firing.



(b) Normally closed valve after firing 1st cartridge to open valve.



(c) Normally closed valve after firing 2nd cartridge to reclose valve.

Figure 35. - Dual-function explosive valve (ref. 68).

A normally open valve is shown in figure 34. In this unit, the explosive charge actuates the ram, which shears through the flow passage and wedges between the openings to seal the flow passage.

A dual-function valve is shown in figure 35 (ref. 68). This valve normally is closed, is opened by firing the first explosive charge, and is reclosed by firing the second explosive charge. Sealing is achieved by interference sliding fits between the lands on the spool and body. A hole is provided to equalize the actuation pressure on each end of the spool subsequent to spool travel.

These examples are typical of some of the explosive-valve configurations. Manufacturers of these valves adapt the style of valve operation to match the system requirements. The usual explosive valve is limited to small line sizes (<1 in. [2.54 cm]); however, by using the energy of the fluid to open and close the valve, with the ram serving only as a latch as shown in figure 36 (ref. 69), larger sizes can be used. The explosive valve is a highly reliable

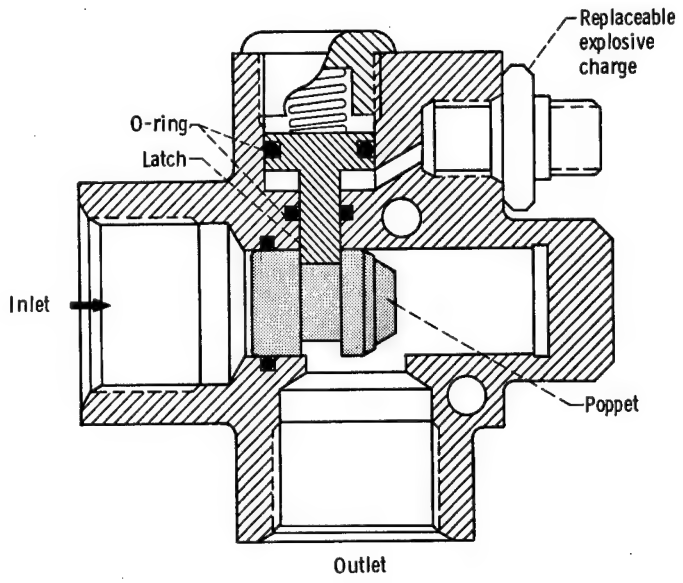


Figure 36. - Reusable explosive-operated valve (ref. 69).

component when properly applied. In critical applications, redundant explosive charges have been used. To increase reliability, the redundancy is applied throughout, with parallel power circuits and parallel rams.

Explosive charges and initiators are vital to the design and proper operation of explosive valves, but discussion of these complex components is outside the scope of this monograph. The treatment of explosive valves herein is restricted to consideration of the valve body, valve actuator mechanisms, and operational problems.

2.5.1 Body

The body of the valve has passages that connect to the inlet and outlet and provide geometry suitable for the operation of the valve. A problem can arise in maintaining a sufficiently low pressure drop through these passages. In some valves, these passages are straight through with minimum change in section, as in figure 34, whereas other valves have right-angle turns, changes in passage size, and impediments to flow such as the ram in figure 33. In some instances, the installation of several valves in a single body with drilled connections has resulted in excessive pressure drop because of discontinuities, sharp-angle turns, and the production of vortices. Opening the passages, blending the intersections, and joining passages so that the centerlines intersect reduces the pressure drop. The ultimate solution is to use a straight-through valve, with attention to the manifolding to preserve the benefits gained from use of this valve.

Some bodies have a critical diaphragm area, as shown in figure 33. The diaphragm must be strong enough to resist burst-pressure forces but not so strong that the explosive charge has to be increased. The material must be compatible with the fluids of the system.

2.5.2 Actuator

The energy source for actuation is provided by an explosive charge or solid propellant housed in a cartridge. Upon firing with an electrical charge, the expanding gases apply force to an actuating ram.

There are two forms of actuation, which differ in the manner in which pressure is developed. In one, the ram that performs the function of opening or closing the valve is actuated by gas pressure from the burning of a solid propellant cartridge that is sealed by an O-ring or similar seal on the ram (fig. 36). In the other, the ram is press-fitted into the valve and an explosive charge is detonated to cause the ram to move (fig. 35(b)). Some leakage of gas can be expected with the O-ring seal before the seal is pressurized and seated.

2.5.3 Operational Problems

The explosive valve, because of the nature of the actuator and the method of opening and closing the valve, presents some problems that are peculiar to this type of valve. These problems include leakage at the connector pins, leakage of actuator gas, seal failure, escape of debris, excessive shock levels, accidental operation of the valve, and effects of ambient temperature.

2.5.3.1 CONNECTOR LEAKAGE

The pyrotechnic charge that moves the ram may develop pressures as high as 60,000 psig (414 MN/m^2). A potential weak spot exists when connector pins are insulated from the body by glass or ceramic seals (potting material). The seals may be glass for low pressure or fused-ceramic for high pressure. Failures to actuate completely have occurred because leaks around the pins or pins that were blown out allowed all the gas to escape without actuating the ram. The problem usually results from a failure to make certain that the coefficients of thermal expansion between the pin and the glass or ceramic potting material are such as to maintain the glass or ceramic in compression; such brittle materials fail very easily if loaded in tension. Epoxy potting material often is used because of its high strength. Stainless steel pins are not used with glass or ceramic seals; one of the high-nickel-iron alloys is used to prevent thermal expansion problems.

2.5.3.2 LEAKAGE OF ACTUATION GAS INTO SYSTEM

The explosive charge of solid propellant cartridge that actuates an explosive valve produces high-pressure gas very rapidly; this gas must be sealed behind the moving ram. When pressure-loaded seals such as O-rings are used, a time interval exists before the seal deflects under the pressure load and establishes a tight seal; during this interval gas can bypass the seal and enter the system. With very reactive propellants such as storable oxidizers, this gas may initiate a reaction; in a hydraulic system, the gas may degrade the stiffness of the system. The use of a press-fit on the ram, as shown in figures 32, 33, and 35, eliminates this problem by providing a highly preloaded seal independent of pressure loading by the gas. Valves without this feature may use a taper on the ram that engages a taper in the body at the end of the stroke (fig. 34) as a means to reduce this leakage.

2.5.3.3 SEALS

Most explosive valves have seals that come in contact with the controlled fluid. These seals may be on the explosive charge (fig. 33), on the inlet fitting (fig. 32), or on the ram (fig. 35). Such seals must be compatible with the controlled fluid. For some propellants, this requirement may be difficult to satisfy; chlorine trifluoride, for example, reacts with all organic materials including Teflon in dynamic situations. This reactivity has resulted in external leakage, destruction of wiring, and fires.

Leakage of fuel and oxidizer together is also hazardous. One solution is the use of pressure-loaded metal seals with a soft metal cladding to adapt to surface irregularities. In this case, the metallic seal materials also must be compatible with the fluid. Another solution is the use of press-fits and welding to replace seals. For example, in the design in figure 33, if the charge container with an already assembled ram were welded into the body and the plug stop were welded in place, the press-fit of the ram in the charge container

remains as the only seal. The valve shown in figure 35 is designed with press-fits as the only fluid seals.

2.5.3.4 ESCAPE OF DEBRIS

Many explosive valves have elements that must be broken to open or close the valve. The element parts must be trapped or retained as shown in figures 32 through 34 to prevent plugging of downstream passages. However, trapping of the gross portion of the failed element does not guarantee that small debris will not be carried downstream. In some instances, regulating valves with a stroke of 0.020 in. (0.51 mm) have been held open by such particulate matter. One solution is a filter that will remove any particulate matter that cannot pass through the valve. Another is the use of a valve configuration that depends on a press-fit for sealing (e.g., the valve in fig. 35) or on an O-ring seal as shown in figure 36.

2.5.3.5 SHOCK

The explosive charge or solid propellant cartridge that actuates an explosive valve produces gas very rapidly, and a very high pressure is reached quickly. In the ignition of an explosive charge, an actual detonation occurs and produces large accelerations and decelerations that may result in high surge pressures. Shock levels of 100 g to 300 g have been measured on structures adjacent to explosive valves at the time of actuation; shocks of this magnitude have damaged sensitive guidance and communication equipment. To avoid excessive shock to the vehicle, the magnitude of the shocks resulting from the operation of an explosive valve must be reduced or the valve must be isolated.

The magnitude of the shock can be reduced by using slower burning charges or less massive rams. Detailed information on the shock levels to be expected with a given valve are not available, since these levels depend greatly on both the mounting and the associated vehicle; testing in the vehicle may be the only way to determine the shock levels. Isolation devices installed between the valve and the sensitive equipment are another way of reducing shock if a particular valve must be used and the shock level is too high.

2.5.3.6 ACCIDENTAL OPERATION

Explosive valves carry their own source of energy in an explosive charge or solid propellant cartridge and hence need only a small amount of electrical energy supplied to the initiator in order to be actuated. Their rapid and predictable response depends on this characteristic. Consequently, explosive valves are vulnerable to stray voltages in the system that may be internally generated by accident, induced by radiation from adjacent equipment, or naturally produced as static charges. High-voltage static charges have been produced by the removal of the valve from a non-conducting plastic bag. The charges can perforate through materials normally considered as insulators and can fire the explosive by the bridgewire-to-case route.

The consequences of accidental operation of an explosive valve depend entirely on the system and can range from a nuisance to an extreme hazard. The solution to the problem lies in a realistic evaluation of the expected operating conditions, good design practices to ensure that equipment is suitable for the intended use, and adequate specifications for assembling, inspecting, handling, testing, and using such equipment.

Guides for the design of electro-explosive devices are presented in references 70 through 72. Guides for the exploding bridgewire firing unit are given in references 73 through 75. A guide to safe practices is presented in reference 76. A recent development in protecting electro-explosive devices from static discharge, presented in reference 77, consists of establishing preferential path circuitry in parallel with the bridgewire, using a small neon lamp and a resistor in series.

2.5.3.7 TEMPERATURE EFFECTS

Ambient temperature can effect the response of explosive actuated valves. Over the normal temperature range (-30°F to +130°F [239 K to 328 K]) only minor effects are noted. However, at temperature extremes (e.g., cryogenic) a major effect on response may occur as a result of a change in ignition rate or burn time of the solid propellant making up the explosive charge. The use of explosive actuated valves may not be precluded by these effects, but the effects must be considered.

3. DESIGN CRITERIA and Recommended Practices

3.1 REGULATORS

3.1.1 Assembly

The regulator configuration shall be as simple as possible consistent with the system requirements.

Use the graph in figure 1 as a guide in evaluating types of regulators. It should be noted that, as the constraints of the system increase, the complexity of the regulator increases. When the system in which the regulator is to be used is not seriously affected by a change to a simpler system, negotiate the specified requirement that will make possible the use of a less complicated assembly.

3.1.1.1 DIRECT ACTING

In a direct-acting regulator, the size of the inlet valve and the characteristics of the reference spring shall maintain the regulated-pressure error within the allowable limits.

Size the inlet valve by the methods described in section 3.1.2. To determine the regulated-pressure error due to variation in supply pressure and flow demand, write the steady-state force-balance equation for the inlet valve, its actuator, and its springs, assuming a direct-acting pressure-unbalanced inlet valve. For the arrangement shown in figure 2, the force balance equation is

$$P_A A_A + L - YX - P_R A_A + P_R A_i - B - ZX - P_S A_i = O \qquad (2)$$

where

A_A = actuator area

A_i = inlet valve area

P_S = supply pressure

P_R = regulated pressure

P_A = atmospheric pressure

X = inlet valve stroke

L = reference-spring installed load

Y = reference-spring rate

B = bias-spring installed load

Z = bias-spring rate

Writing this equation in differential form, the regulated-pressure error ΔP_R may be expressed as

$$\Delta P_R = -\frac{\Delta P_S A_i}{A_A - A_i} - \frac{\Delta X(Y+Z)}{A_A - A_i} \tag{3}$$

where the atmospheric pressure P_A is zero.

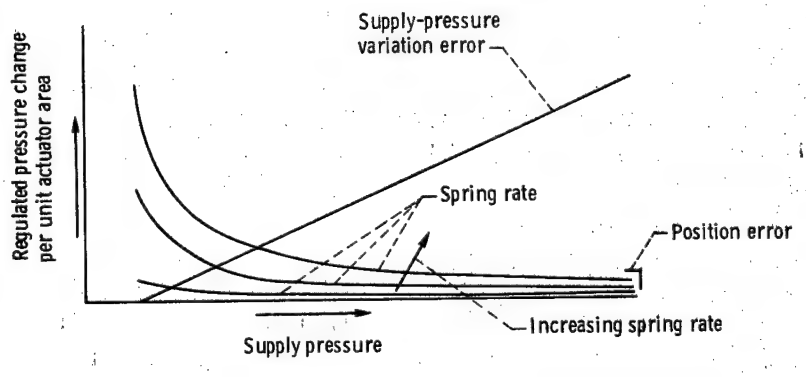
The first term on the right side of equation (3) is the error due to supply-pressure variation, and the second term is the error due to valve position or flow demand. Determine both errors per unit actuator area as a function of supply-pressure variation. A typical plot is shown in figure 37(a), with the sum of the two shown in figure 37(b) (note that the regulated-pressure change in these figures is negative). Size the actuator and spring to limit ΔP_R to approximately 25 percent of total regulated-pressure tolerance. Select a spring rate that will result in the minimum net change of total error as shown in figure 37(b) (ref. 15).

It should be noted that the error due to supply-pressure variation will decrease with decreasing inlet valve area and will be minimum at the minimum possible inlet valve area. However, the inlet valve stroke will increase as the inlet valve area decreases. The greater valve-position error resulting from this stroke increase can be reduced by a decrease in spring rate. When the inlet valve area exceeds the minimum possible area, the design should be studied for possible improvements. Adjust the regulated-pressure set point with the supply pressure at midpoint of the range.

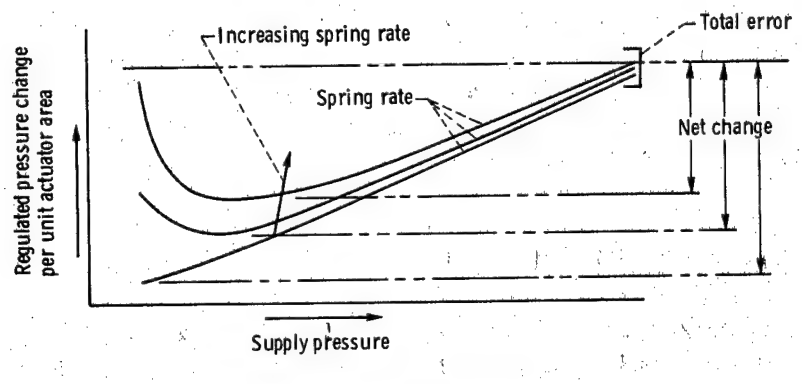
3.1.1.2 DOME LOADED

In a dome-loaded regulator, the size of the loader valve and the characteristics of its reference spring and the size of the inlet valve and the characteristics of its bias spring shall maintain the regulated-pressure error within allowable limits.

Size the inlet valve by the method in section 3.1.2; then follow the procedure described in section 3.1.1.1 to determine the regulated-pressure error of the loader valve due to supply-pressure variation and required loader flow. Because the loader valve seat, actuator, and reference spring are relatively small, the regulated-pressure error will be small.



(a) Individual regulation errors.



(b) Combined regulation errors.

Figure 37. - Regulation-error characteristics of a direct-acting regulator.

Using the error characteristics of the main inlet valve, size the main inlet valve actuator and bias spring to hold the regulated-pressure error to a minimum. Add the two results to determine the total regulated-pressure error due to supply-pressure variation and flow demand. Apply the same limitations to the total error, and adjust the set point as described in section 3.1.1.1.

3.1.1.3 PILOTED

In a piloted regulator, the response of the pilot circuit and the size of the inlet valve actuator and the characteristics of its spring shall maintain the regulated-pressure error within allowable limits.

Using the methods described in section 3.1.1.1, size the actuator to minimize the regulated-pressure error due to supply-pressure variation and flow demand. In a similar manner, size the pilot valve; in this case the supply pressure is the pressure supplied to the pilot (i.e., control pressure), and the flow demand is the bleed flow through the pilot valve. Conduct a steady-state analysis (refs. 19 and 20) to size the various elements of the regulator.

Conduct a dynamic analysis (refs. 21 and 22) to demonstrate satisfactory operation. Consider the following factors that affect operation: sustained acceleration, temperature variation, vibration, mechanical hysteresis, and friction. Evaluate the effect on response of variations in flow orifices and sensed-pressure variables. The bleed flowrate can be an important factor in stable operation in vibration environments; in particular, therefore, study the effects of varying the bleed flowrate.

3.1.1.4 NONMODULATING REGULATION SYSTEM

The pressure-control elements in a nonmodulating regulation system shall have the response time and reliability necessary to maintain the regulated pressure within the allowable limits.

Establish all the design parameters of the regulated-pressure system. Establish the operating characteristics of the pressure-sensing element or pressure switch. Size the shutoff valve to provide the required flow of pressurizing gas. If the response time of a single shutoff valve is too great to meet the required pressure-regulation tolerance, use multiple smaller valves with decreased response time. For increased reliability, use redundant pressure switches in a voting circuit in which two of three pressure switches must actuate to signal valve operation. Perform a dynamic analysis of the total system in its operating environment to prove its adequacy (ref. 23). Consider any tendency for the pressure switches to change setting with usage.

3.1.2 Inlet Valve

3.1.2.1 SIZE AND STROKE

The inlet valve size and stroke shall provide flow area adequate for all operating conditions.

Determine the equivalent orifice size of the inlet valve from the equations for isentropic compressible flow for orifices and nozzles (ref. 25). Base the calculation on the required flowrate, minimum absolute supply pressure, maximum absolute regulated pressure, and maximum operating temperature; use an orifice discharge coefficient of 0.6 unless a different value is known to be appropriate. If the most adverse conditions do not all occur simultaneously, size the valve for the worst combination of actual operating conditions. In preliminary analysis where the final configuration has not been selected, use a flow area 10 percent greater than calculated to include possible added inlet valve losses. For poppet valves, determine the seat diameter and stroke to give an equivalent flow area exclusive of the actuation means. When this method of sizing is used, make the inlet valve the principal flow restriction. For butterfly valves, size the opening to give the net required flow area.

3.1.2.2 FLOW PASSAGES

The flow passages of the inlet valve shall provide the desired flow characteristics.

Size the flow passages and possible inlet filter for as small a pressure drop as possible. Size the flow passage downstream of the inlet valve to ensure subsonic flow for all operating conditions. Make the inlet valve the principal flow restriction.

3.1.2.3 PRESSURE BALANCING

A pressure-balanced inlet valve shall provide narrow-band regulation and stable operation.

Choose the type of pressure-balanced inlet valve suitable for the application according to the following guidelines:

(1) Use a dynamic seal on the balancing piston of the inlet valve when (a) the seal and its lubricant, if required, are effective over the operating temperature range and (b) the regulated-pressure error resulting from the hysteresis and friction caused by the seal does not exceed 25 percent of the allowable regulated-pressure tolerance.

(2) Use a double-seated inlet poppet or butterfly when positive shutoff is not required. Size the double-seated poppet with one-half the flow going through each seat.

(3) Use a piloted inlet valve when (a) a positive shutoff is required, (b) the permissible error due to supply-pressure variation and flow demand is small, and (c) the regulator size may be reduced appreciably. For stable operation, the piloted open-loop pressure-balanced inlet valve is preferred over the closed-loop inlet valve.

3.1.3 Actuator

Complete details on recommended design practices for actuators may be found in reference 31. The following criteria and practices are specific to actuators for regulators.

3.1.3.1 TYPE

The actuator type shall be suitable for the application and shall satisfy the regulation requirements.

Use pistons with dynamic O-ring seals where the operating temperature range is limited to -65° to +160°F (219 to 344 K), the fluids are compatible, and the operation will not be affected adversely by friction. Use diaphragms where strokes are relatively short and diameters are large, temperature variations are extreme, compatibility with fluids is critical, and friction will affect operation adversely. Use metallic diaphragms such as Inconel 718 or beryllium copper wherever possible for critical applications when control of a material properties is important. Limit the use of Mylar, Teflon, elastomeric materials, nylon, and fiberglass to non-critical applications. Use bellows where the inlet valve requires a large stroke.

3.1.3.2 SIZE

The actuator shall provide sufficient force to operate the inlet valve as required to control the regulated pressure within allowable limits.

Establish the actuation-pressure variations that will permit operation within the allowable regulated-pressure tolerance, response time, bleed flow, and controller operation. Determine required force output of the actuator. Determine actuator size (ref. 19). Use a pressure-balanced inlet valve to reduce actuator size if the valve does not degrade overall regulator operation.

3.1.3.3 RESPONSE

The actuator response to pressure variations shall not induce sustained instability or out-of-tolerance operation.

Using computer techniques (refs. 21 and 32), perform a dynamic analysis of the effects of variations in the damping-fluid volumes around the actuator and in the orifices metering the flow of the damping fluid into these volumes. Select the combination of volumes and orifices that optimizes operation. Modify the orifices as necessary for different gases or mixtures of gases.

To correct for high-frequency noise, increase volumes and decrease orifice sizes. To correct for transient overshoot or undershoot of regulated pressure, decrease volumes and increase orifice sizes.

3.1.3.4 BIAS SPRING

When a dome-loaded regulator is operated at minimum supply pressure and maximum flow demand, the closing force of the actuator bias spring shall allow the inlet valve to open sufficiently to maintain minimum regulated pressure.

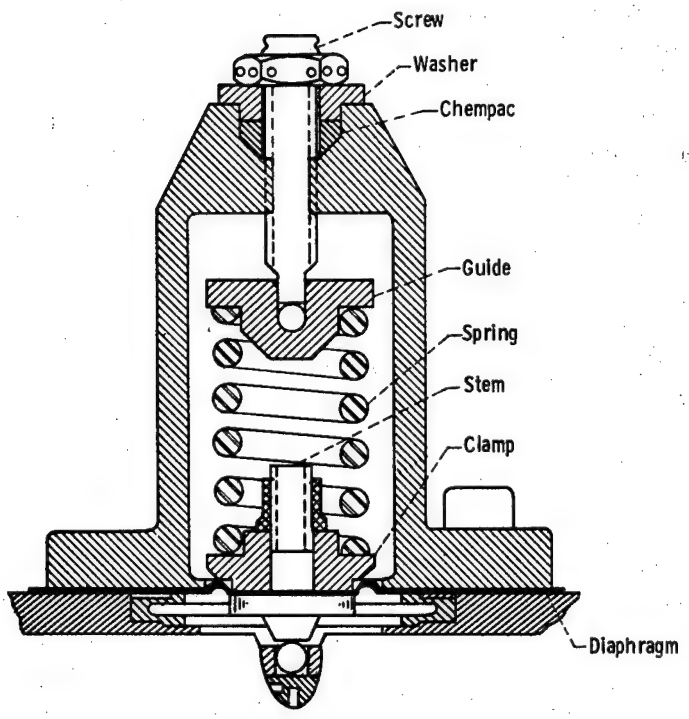
Use a variable-rate bias spring. Use the portion of the load-vs-stroke curve where the spring force decreases with increasing stroke. A Belleville spring has this characteristic; a Belleville bias spring and actuator are shown in figure 3.

3.1.4 Control Elements

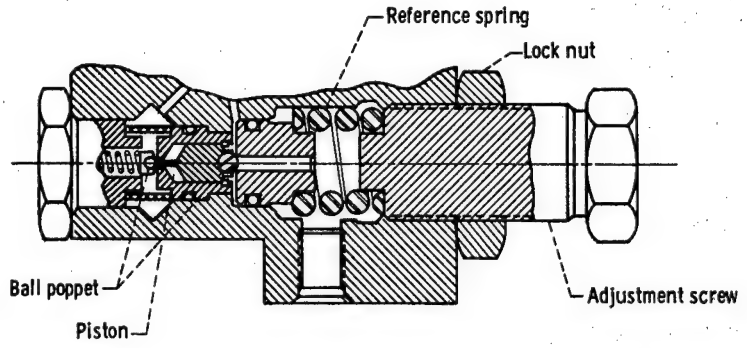
3.1.4.1 SPRING REFERENCE LOAD

The regulated-pressure error resulting from the spring reference load shall be as small as possible.

Use a spring support that provides a degree of freedom to reduce side loads on the actuating mechanism; mechanical arrangements should have as little friction as possible. Spring supports should be similar to those shown in figure 38. When the squareness of the reference-spring end is critical, check the spring in the loaded condition in a fixture similar to the one shown in figure 39. Rotate the spring to determine the deviation from normal to axis of rotation. Grind the spring ends until the desired squareness under load is obtained.



(a) Spring support, diaphragm sensor.



(b) Spring support, piston sensor.

Figure 38. - Reference-spring installation.

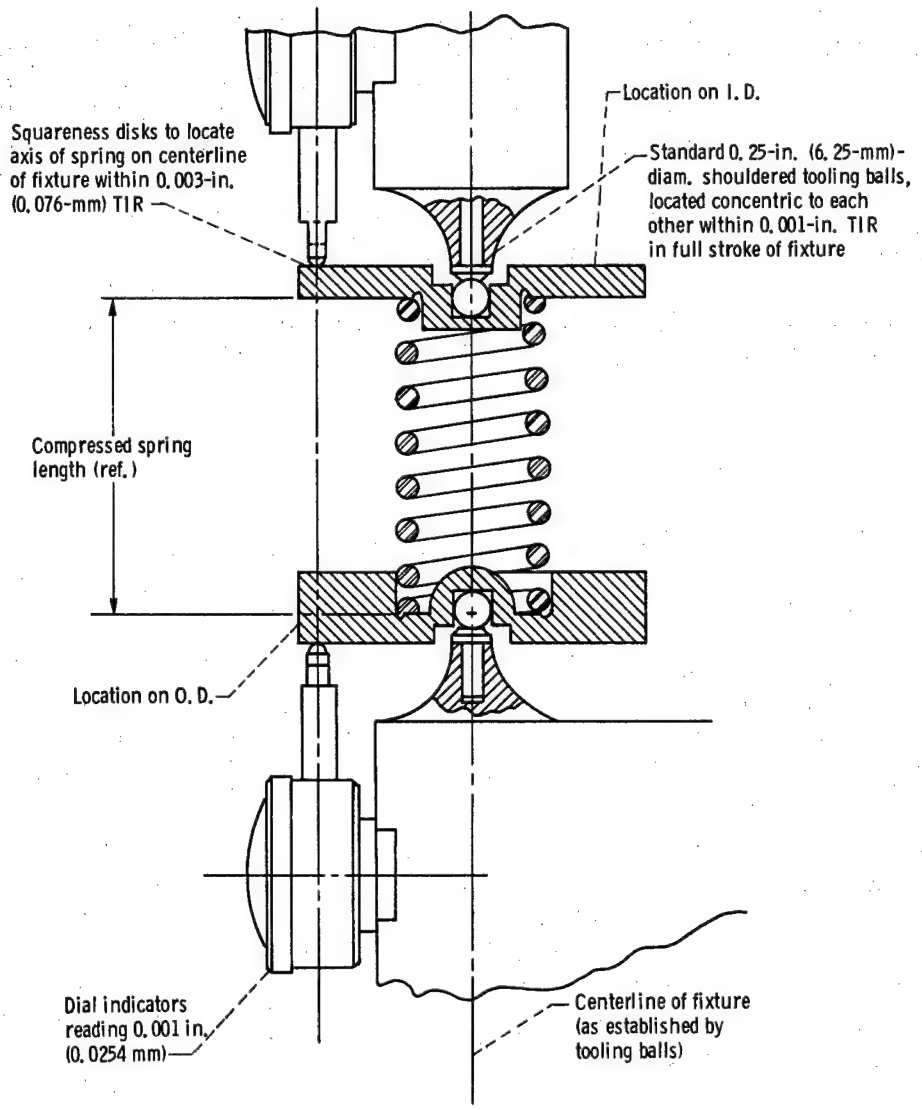

Location on I. D.
Squareness disks to locate axis of spring on centerline of fixture within 0. 003-in. (0. 076-mm) TIR
Standard 0. 25-in. (6. 25-mm)-diam. shouldered tooling balls, located concentric to each other within 0. 001-in. TIR in full stroke of fixture
Compressed spring length (ref.)
Location on O. D.
Dial indicators reading 0. 001 in. (0. 0254 mm)
Centerline of fixture (as established by tooling balls)

Figure 39. - Test fixture for checking squareness of spring ends.

3.1.4.2 PILOT VALVE CIRCUIT

The pilot circuit shall supply the desired control to the regulator and adapt to the physical arrangement of the regulator.

List the performance characteristics that the piloted regulator assembly must meet. To meet these requirements, select the best pilot circuit from the schematics shown in figures 8 and 9. Use zero-gain pilot circuits for dome-loaded regulators, positive-gain pilot circuits for back-pressure regulators, and negative-gain pilot circuits for pressure-reducing regulators. If a regulator must lock up, select a pilot circuit that locks up and has a restriction inlet. If a regulator permits a continuous external bleed, choose a pilot circuit that bleeds and has a restriction inlet or a pilot circuit that locks up or bleeds with a restriction outlet. If a regulated system is critical to a regulated-pressure increase that occurs when flow is terminated suddenly, provide a restriction inlet. If a regulated system is critical to a regulated-pressure decrease that occurs when there is a sudden increase of flow demand, provide a restriction outlet. When several pilot circuits are satisfactory, select the circuit that adapts best to the regulator configuration.

3.1.4.3 ORIFICE

Orifice geometry and installation shall provide effective flow control.

Size the orifice from the equation for isentropic compressible flow for orifices (ref. 25). Determine the orifice diameter for the desired flow at the anticipated operating conditions; use an orifice flow coefficient consistent with a sharp-edge hole flowing the specified gas. Use sharp edges with 0.001-in. (0.025 mm) break or less. If an orifice plate is used, make the thickness of the plate less than 1/8 of the orifice diameter or 1/30 of the flow-passage diameter, whichever is smaller. Make both sides of the plate identical so that it may be installed either way. If both sides of the orifice cannot be made sufficiently identical for flow in either direction, design the orifice plate to prevent backward installation. For extremely accurate flow control, flow-calibrate the orifice and rework it, if necessary, to meet the required flow capacity. Provide special filter screens for critical small-diameter orifices. Seal orifice plates in flow passages to prevent bypass flow or leakage. Do not use sintered metal for flow control.

3.1.4.4 THERMAL COMPENSATOR

The regulated-pressure set point shall not shift when the regulator is exposed to temperature variations.

Use Ni-Span-C wire for the reference-spring material. Use a combination of Invar and CRES to limit the variation of reference-spring load resulting from thermal expansion or contraction of the supporting structure. Teflon, Kel-F, or nylon should not be used in

thermal compensators. Satisfactory designs for thermal compensators are shown in figure 10. Protect the Invar from corrosion; if the Invar is plated with electroless nickel, ensure that possible flaking of the nickel will not contribute to a malfunction of the assembly.

3.1.4.5 SHUTOFF VALVE

A piloted regulator used as a supply-pressure shutoff shall maintain the inlet valve in the closed position.

Use the main inlet valve of the regulator as the primary system shutoff and use an electrically operated shutoff pilot valve in the low-pressure control circuit. In the deenergized position of the shutoff pilot valve, ensure that the regulator inlet valve is closed. If the system downstream from the regulator is dead-ended without leakage, use a three-way shutoff pilot valve with a vent port to exhaust leakage occurring in the normally deenergized position. Provide a low-flow-capacity relief valve upstream of the shutoff pilot valve to prevent overpressurization of the low-pressure control circuit of the regulator.

3.1.4.6 FLOW LIMITER

The regulator shall not permit a rate of flow that results in overpressurization of the regulated system.

Add a flow limiter to any regulator that permits a high flowrate and system overpressurization. Use a design with demonstrated high reliability, e.g., one of those shown in figure 11.

3.1.5 Operational Problems

3.1.5.1 CONTAMINATION

The regulator shall not contain contaminant that can damage critical surfaces or cause malfunction.

Provide a filter in the inlet of the regulator. Size the filter for minimal pressure drop, with no increase in pressure drop when the filter area is 25-percent clogged with the contaminant to be removed. Choose the degree of filtration (pore size) to protect critical elements. In cryogenic systems where "snow" is a problem, use a filter having an absolute rating of 10μm. Establish procedures to ensure the cleanliness of detail parts and assemblies during fabrication, assembly, and storage.

3.1.5.2 LUBRICANTS

Lubrication of the moving elements of a regulator shall not induce erratic operation.

Avoid the use of lubricants in regulators for cryogenic service. When galling due to high loads occurs, use a dry-film lubricant selected on the basis of the guidelines in reference 49. In regulators with dynamic O-rings, lubricate with silicone grease such as Dow Corning DC-55 or FS 1281; but do not use FS 1281 at temperatures below 0°F (256 K). Establish procedures to control the amount of lubricant and method of application. Reference 48 provides guides for the selection of liquid lubricants.

3.1.5.3 VIBRATION

In a bleed-type pilot-operated regulator, the regulation error produced by vibration-induced motion of the controller valve shall be as small as possible.

Provide stops on both extremes of the controller valve travel. Determine the controller-valve full-open flow at the required controller inlet and outlet pressures. Adjust the bleed flow to one-half the measured controller valve full-open flow. This adjustment will position the controller valve at midposition between the open and closed position stops at nominal steady-state operation and will minimize the regulated-pressure error caused by vibration.

3.1.5.4 MOISTURE IN REFERENCE CAVITY

The regulator design shall preclude any form of liquid in the atmospheric-pressure reference cavity.

Use a cover that has a series of flow paths arranged in such a manner that fluid entering an opening in the cover will flow out another opening in the cover and not into the reference cavity. A proven design is shown in figure 12. Sliding fits within the reference cavity should have a minimum overlap to reduce the susceptibility to malfunction due to freezing. Establish methods of packaging and procedures for drying that ensure that cavities are kept free of moisture during storage and during periods between operation in the space vehicle.

3.1.5.5 STABILITY

A regulator shall not exhibit any form of instability caused by flow dynamics or interaction with other system components.

Locate the internal controller sense point where flow turbulence is at a minimum. External sensing should be located as close to the regulator as possible; avoid long transmission lines.

To ensure stable operation of the regulator and all other components, conduct a system dynamic analysis by means of a digital computer program. Analysis should be performed only by highly qualified personnel familiar with regulator operation.

3.1.6 Leak Testing

Tests shall verify that the leakage of a regulator in a lockup mode of operation does not exceed the allowable limit.

Use the following procedure for ensuring that a regulator in a lockup mode is not leaking to the downstream system more than the specified allowable amount:

(1) Check the test setup shown in figure 40 for external leakage. The leakage should be zero when checked with non-corrosive leak-test compound.

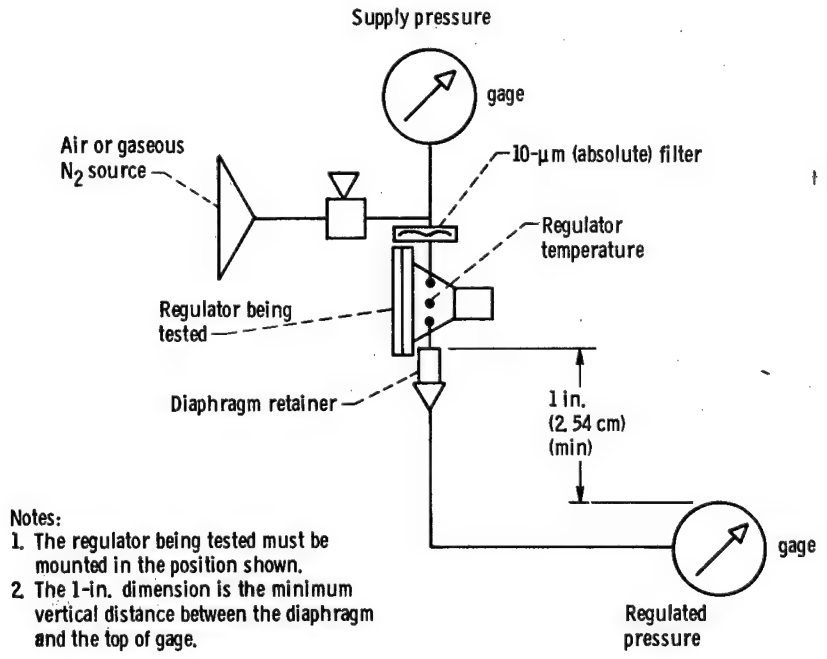

Supply pressure
gage
Air or gaseous N_2 source
10-μm (absolute) filter
Regulator temperature
Regulator being tested
Diaphragm retainer
1 in. (2.54 cm) (min)
gage
Regulated pressure
Notes:
1. The regulator being tested must be mounted in the position shown.
2. The 1-in. dimension is the minimum vertical distance between the diaphragm and the top of gage.

Figure 40. - Schematic of test setup for determining regulator leakage.

(2) Fill the regulated-pressure gage and system to diaphragm installation surface with hydraulic oil. This portion of the system should be completely void of entrapped gas.

(3) Install an isolation diaphragm in the diaphragm retainer. Complete test setup.

(4) Pressurize the air or gaseous nitrogen to the maximum supply-pressure requirement of the regulator.

(5) Record the initial temperature and pressure of the system downstream of the regulator. This pressure level shall be within the allowable lockup-pressure range of the regulator.

(6) Hold for the designated time period of the test (usually a number of hours).

(7) Record the final temperature and pressure.

(8) Substitute the initial and final temperatures and pressures in the equation

$$\text{Leakage} = \frac{C}{T_1+T_2}\left[\frac{P_2-P_1}{t} - \left(\frac{P_1+P_2}{T_1+T_2}\right)\left(\frac{T_2-T_1}{t}\right)\right] \qquad (4)$$

where

T_1 = initial temperature

T_2 = final temperature

P_1 = initial pressure

P_2 = final pressure

t = length of test

$C = \frac{2V}{\gamma GF}$

V = volume of gas downstream of regulator inlet valve

γ = ratio of gas specific heat at constant pressure to specific heat at constant volume

G = specific gas constant

F = conversion factor for weight flow to volume flow

(9) Repeat the test with the pressure applied to the regulator equal to the minimum supply-pressure requirement of the regulator.

Determine the constant C from equation 4(a), using the maximum anticipated volume V and the minimum values for specific heat ratio γ and specific gas constant G for gas used in the test. Modify the constant C for pressure- and temperature-measuring instrument errors, using the combination that results in the largest leakage value.

3.2 RELIEF VALVES

3.2.1 Size

The relief valve shall provide adequate flow area under all operating conditions.

Using the equation for isentropic compressible flow through an orifice (ref. 25), determine the orifice size equivalent to the required relieving flow area. In the calculation, use the full-flow rate, the absolute inlet pressure at full flow, the maximum operating temperature, and an orifice discharge coefficient of 0.6. Size the valve for this equivalent orifice area. If the relief valve discharges into ducting and a back pressure exists at the valve outlet, use the absolute outlet pressure when determining the equivalent orifice size for conditions of subsonic flow. If a regulator is in the system protected by the relief valve, verify that this flow area is adequate to handle the discharge of a runaway regulator at the most adverse conditions.

3.2.2 Assembly

3.2.2.1 DIRECT OPERATING

The direct-operating relief valve shall have adequate seat size and stroke with a spring as small as possible.

To evaluate the acceptability of a direct-operating (pressure-unbalanced-to-open) valve, prepare characteristic curves similar to those shown in figure 16. From the flow area of the equivalent orifice, determine the seat diameter, reference-spring installed load and rate, and the minimum spring weight as a function of full-flow stroke. In pneumatic flow, limit the curves to the region where the full-flow stroke is less than one-fourth the seat diameter. The curves indicate that the most efficient design (lightweight and small size) is one using the maximum allowable stroke. In actual practice, use a stroke less than the maximum allowable. Units sized to operate in the high-spring-rate region are difficult to adjust to meet the relief-pressure operating range. In designs where the full-flow stroke is limited, consider an approach other than the direct-operating, unbalanced-to-open type. When flow requirements are large and the spring becomes excessively large, consider the use of a piloted-type valve.

For the nonmodulating relief valve shown in figure 13(b), select a seat diameter and poppet stroke for which a Belleville spring may be sized with the desired reference load and stroke in the negative-spring-rate range of the Belleville load-stroke curve. Use as short a stroke as possible, because the Belleville spring has a short stroke; provide an open-position stop to limit the stroke. Provide a pressure-balance passage to maintain equal pressure on both sides of the Belleville spring to prevent an actuator-type force being applied to the poppet.

3.2.2.2 INVERTED OPERATION

The inverted-operation relief valve shall meet flow requirements and at reseat pressure shall provide a poppet-to-seat load as large as possible.

To evaluate the inverted-operation, pressure-unbalanced-to-close valve, prepare characteristic curves similar to those shown in figure 17. In this valve, the pressure at which the hanger contacts the poppet and starts to lower the pressure-unbalanced closing forces should equal the nominal operating pressure or the defined reseat pressure. The reference-spring installed load will vary with the pressure-unbalanced-to-open force resulting from the effective diaphragm area. Using the flow area of the equivalent orifice, determine the seat diameter, diaphragm diameter, reference-spring installed load and rate, and the minimum spring weight as a function of full-flow stroke. In pneumatic flow, limit the curves to the region where the full-flow stroke is less than one-fourth the seat diameter. Although the curves indicate that the most efficient design is at the maximum allowable stroke, use a stroke less than the maximum allowable. When flow requirements are large and relieving pressures are high, the spring becomes excessively large; consider the use of a piloted-type valve.

3.2.2.3 PILOT OPERATED

The pilot-operated relief valve shall satisfy full-flow and reseat requirements and shall be as small as possible.

To select a piloted-type relief valve, prepare schematics of various configurations. Perform steady-state analysis of these configurations (ref. 51). Use the arrangement that will meet the operational requirements and fit the allowable envelope. To aid in the selection of valve operating characteristics, prepare curves similar to those shown in figure 18. Use a plot like that in figure 18(a) to select the stroke of the pilot valve at which the actuator control pressure permits the main poppet to start to open. Use the difference between the cracking pressures of the pilot valve and main valve to determine the reference-spring characteristics as shown in figure 18(b). The seat loadings of the pilot poppet and main poppet are established as shown in figure 18(c). Conduct a dynamic analysis to demonstrate response and stability characteristics. Modify valve sizing as necessary.

3.2.3 Inlet Valve

3.2.3.1 POPPET AND SEAT

The inlet valve poppet and seat shall prevent leakage and have repeatable relieving characteristics.

For direct- or inverted-operating relief valves or pilot valves of pilot-controlled relief valves, use poppet and seat material that is hard metal-to-metal (heat-treated 440 C or equivalent) with a 1-μ in. (0.0254 μm) AA surface finish in the sealing areas and a minimum seat loading stress of 1000 psi (6.895 MN/m^2). Soft materials may be used for sealing the main inlet valve of pilot-controlled relief valves. To achieve lower costs for manufacture and inspection, use flat poppets and seats.

3.2.3.2 SUBASSEMBLY CONSTRUCTION

The subassembly shall withstand forces that cause changes in cracking pressure.

Use single-piece construction and avoid joints that may separate. A recommended design is shown in figure 19(b). Establish assembly procedures and tooling to prevent injury to personnel performing assembly operation.

3.2.4 Actuator

All the design criteria and recommended practices in section 3.1.3 are applicable.

3.2.5 Control Elements

All the design criteria and recommended practices in sections 3.1.4.1, 3.1.4.3, and 3.1.4.4 are applicable.

3.2.6 Operational Problems

3.2.6.1 MOISTURE IN THE ACTUATOR

The actuator shall be moisture-free, and moisture shall not be introduced by the actuation gas.

Before storage or use, dry the unit for 30 minutes minimum in a chamber at an absolute pressure of 1.0 in. Hg (3.38 kN/m^2) maximum and a temperature of 120° to 130°F (322 to 328 K). Provide a procedure to verify and ensure a dew point for the actuating gas of -79°F (212 K) at 14.7 psia (101 kN/m^2).

3.2.6.2 PILOT VALVE PRESSURE SENSING

3.2.6.2.1 Sensor Location

The portions of the pilot valve that are intended to respond to reference pressures shall be exposed to pressures that vary in a known way with the reference pressures.

Locate the pilot valve sensing inlet at a position in the valve that is at the total pressure. Locate the exhaust of the pilot valve in a position where static pressure approximates atmospheric pressure. This location should not be exposed to pressure ram effects resulting from change in direction of flow.

3.2.6.2.2 Sensor Material

The sensor material shall be compatible with the fluid and the operating conditions to which it is exposed.

When thin sheet material is exposed to hydrogen at elevated pressures and cryogenic temperatures, use Inconel 718. Do not use 17-7 PH stainless in any application that exposes it to hydrogen.

3.2.6.3 STABILITY

The interaction of coupled pneumatic components shall not generate sustained pressure and flow oscillations.

When a combination of pneumatic units is unstable or considered potentially unstable, prepare a computerized mathematical model of the units. The model should represent as closely as possible the volumes, spring forces, valve strokes and size, actuator areas and masses, friction, and pressure transients in the various volumes. Determine the features of the design that contribute to the instability. Vary each feature or combination of features in a series of computer runs to determine the best solution. In particular, check the effects of the following variations in parameters:

- Changes in volume.
- Changes in spring installed load and rate.

- Changes in mass of moving parts.
- Changes in friction of sliding parts.
- Changes in flow capacity of valves, flow passages, and orifices.

Verify the computer solution by tests on a modified unit or on the first unit built. On the basis of values for the parameters measured in the tests, modify the mathematical model to reflect accurately component behavior, in case future problems may need analysis.

3.3 CHECK VALVES

3.3.1 Poppet

3.3.1.1 BALL

The ball and seat in a ball check valve shall withstand the closing impact without allowing excessive leakage.

Limit the use of ball check valves to low-flow applications where the stroke is short and the ball cannot misalign and damage the seat when stroking closed. Do not use ball check valves in applications where the valve may chatter.

3.3.1.2 GUIDED POPPET

The guided poppet shall contact the seat with a continuous and uniform contact.

Establish the guide clearances, length of guiding surface, and normality of the sealing area of the poppet such that the poppet contacts the seat without gaps. Control the width of the sealing area of the poppet and seat to give the desired seat loads. Use hard surfaces on the poppet guide and in the guiding bore; use only hard anodize on aluminum alloys. Leading-edge radius of the guide should blend in tangent with the diameter

3.3.1.2.1 Restrictor

The restrictor check valve shall limit flow in the check direction.

Provide an orifice in the poppet (fig. 22) to limit the flow in the check direction. Provide a means to seal off the orifice while measuring the poppet-to-seat leakage. Provide a filter in the poppet to protect small-diameter orifices from being plugged by contaminants.

3.3.2 Flapper

The flapper check valve shall withstand operating temperatures and fluid flow forces, and the pressure drop in the valve shall be as small as possible.

Protect the torsion spring from the flowing fluids. Size the torsion spring for operation at a low level of unit stress at the highest operating temperature. Stress the flappers to resist the high impact loads that occur when the flappers strike the open-position stops. Control the open position of the flappers to prevent rattling. Use split-flapper check valves when the envelope is limited.

3.3.3 Operational Problems

3.3.3.1 INSTALLATION

The check valve design shall preclude incorrect installation.

Identify the direction of flow on the valve body with a permanently marked, easily seen arrow with the note "DIRECTION OF FLOW." Make the connectors on each end of the check valve different to ensure a single mode of installation with the correct direction of flow. Do not use check valves with identical end connectors.

3.3.3.2 CHATTER

Exposure to low-flow conditions shall not result in check valve chatter or repeated actuation that may exceed the normal actuated life.

When the poppet check valve is sized, provide a flow-induced opening force that will hold the poppet off the seat during low-flow modes of operation. Use a return spring with a low spring rate. Figure 41 illustrates a design with minimal tendency to chatter. If it is not possible to design for chatter-free operation, limit the operating time at the chatter conditions to less than the normal operating life of the valve. Use hard bearing surfaces on the sliding members. Hard anodize the aluminum bearing surfaces.

If a flapper valve chatters, analyze the operating condition inducing the chatter. Revise the design accordingly, as illustrated in section 2.3.3.2. When a design revision is not feasible, limit the exposure time at the chatter conditions as noted above.

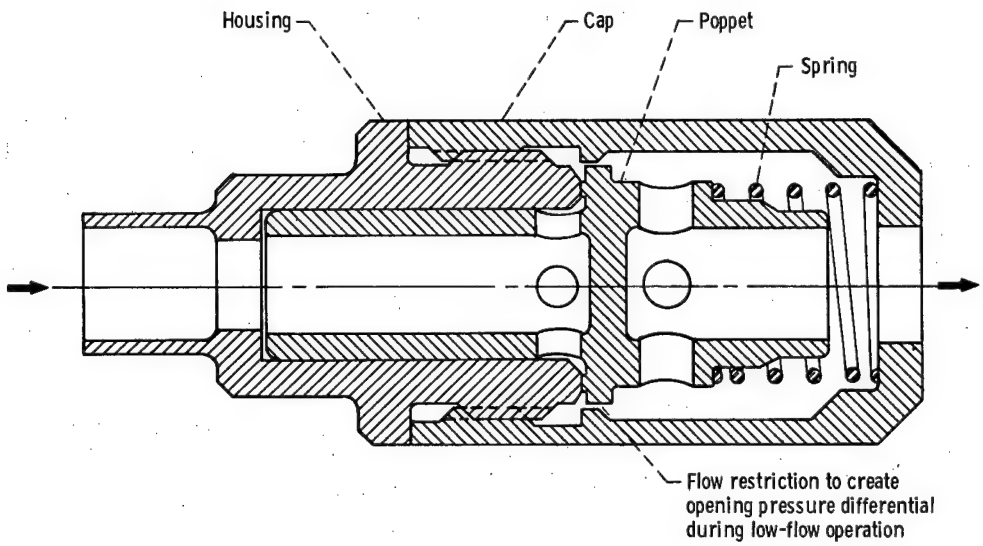


Figure 41. - Check valve design to reduce chatter.

3.3.3.3 POPPET CLOSING TIME

The check valve poppet shall close rapidly enough to prevent reverse flow.

Review the check valve design for any operational feature that could retard the motion of the checking element. A feature that may allow a seal between the poppet and the open-position stop is particularly undesirable.

3.3.3.4 CONTAMINATION

Contaminants in the system fluid shall not impair operation of the check valve.

To minimize the effect of contaminants in the flowing fluid, provide filters upstream of the inlet to the check valve. (Do not provide filtration to both ports of the valve, as assembly-introduced or self-generated contaminants will be trapped in the valve and accelerate the damage resulting from contamination). Size the filter so that the anticipated clogging from contaminant particles (including snow) will not increase the filter pressure drop even with 25 percent of the filter surface area blocked.

Provide dual check valves installed in series in critical applications where check valve failure could cause engine system or component damage and result in mission failure. Design a poppet and seat construction that will withstand maximum anticipated impact loads.

3.3.3.5 VENTING

Check valves installed to vent a cavity and prevent entrance of elements of an adverse environment shall not malfunction when exposed to that environment.

Make sliding or moving parts of the check valve internal to the poppet seal. Use an external poppet configuration that will minimize the increase of cracking pressure that results from moisture freezing on the outside of the assembly. A configuration that loads frozen moisture in tension is recommended. Avoid using poppets enclosed in recessed cavities. Rubber compounds compatible with the system fluids should be used for seats, poppets, and seals. These compounds will seal adequately at relatively low temperatures and prevent the entrance of snow. Control the flatness and finish of the rubber surface to ensure sealing when the rubber is hard. Attempt to achieve a sealing surface approaching that of hard metal-to-metal poppets and seats. To protect assemblies, install the valves in areas where they are shielded from handling damage. Use a rugged construction in the sealing area. Use figures 24(a) and (b) as guides to acceptable design.

3.4 BURST DISKS

3.4.1 Burst-Pressure Capability

The burst disk shall rupture reliably at the required pressure.

Use a disk with a burst pressure that will permit initiation of system operation at the lowest acceptable pressure level, yet will not rupture prematurely as a result of expansion of fluid or changes in vapor pressure, tolerances, or physical properties of materials with temperature and time. When burst disks are used in a pressure-sequence system, use a burst pressure that will permit all necessary prior operations to be complete before the disk ruptures. In the burst-pressure sample testing, test at the same rate of pressure rise as in the application.

Use a shear-type burst disk as shown in figure 26 for applications in which the burst pressure must be set within a narrow tolerance band. Set the Belleville load to compensate for temperature and material variations. Load the Belleville spring in the negative-spring-rate region of the stroke. Select materials that will not deteriorate with time. Protect downstream components by providing a catch screen for the sheared disk.

3.4.2 Reverse-Pressure Capability

The disk shall withstand reverse pressure without rupture.

In the case of prebulged disks that are exposed to vacuum on the upstream side and atmospheric pressure on the downstream side, support the disk if the burst pressure is less than 1000 psid (6.895 MN/m^2). Support the flat coined-groove disk if the reverse pressure is greater than 50 percent of the rated burst pressure, unless the reverse-pressure capability is known accurately. Design the support structure so that it does not impede the flow when the disk ruptures.

3.4.3 Size and Thickness

The size and thickness of the burst disk shall ease the criticality of other design parameters.

The decision to use a burst disk in a fluid system should be made early in the system design, so that maximum freedom in the choice of diameter and thickness of the disk is provided. Then make the disk large enough to (1) avoid close manufacturing tolerances and (2) minimize pressure drop after rupture. Keep in mind that diameter and thickness are interrelated and are functions of material properties (sec. 3.4.5). In general, when possible, use aluminum alloys; these materials allow more latitude in choice of size and thickness than can be achieved with other materials.

3.4.4 Coined-Groove Pattern

The coined-groove pattern shall satisfy the requirements for pressure drop, flow disturbance, and reverse-pressure capability.

Use the radial coined-groove burst disk when a relatively high pressure drop is allowable, as in pyrotechnic applications. Use the single-hinge type when the maximum ratio of reverse pressure to burst pressure is desired. Use the double-hinge type when the lowest tolerance on burst pressure is desired. Make the thickness remaining under the coined groove approximately 10 percent of the material thickness.

3.4.4.1 PETAL RETENTION

The petals shall be retained after bursting unless their passage downstream can be tolerated.

Make the hinge sufficiently wide to resist failure due to flow forces and temperature variations. When necessary, provide a chamfered notch (fig. 27) to prevent tearing of the hinge. If possible, face the coined groove upstream. When the petal cannot be retained, provide a catch screen.

3.4.5 Material

The burst disk material shall be suitable for the application.

Use weaker materials, such as aluminum, for relatively low burst pressures and small disks. Limit the use of aluminum to operating temperatures less than 250°F (394 K). Choose stronger materials such as Inconel for high burst pressures and large diameters. Where zero external leakage is required, select materials that may be welded to the holder so that the strength of the burst disk is not lowered. Use materials that will not corrode in the contained fluid and that will not react chemically with the contained medium (e.g., avoid aluminum with hydrazine-based monopropellants). When the burst disk is clamped, choose materials that do not promote galvanic corrosion (ref. 78). Use nonmetallic coatings such as Teflon and Kel-F for corrosion protection, provided that the temperature is less than 450°F (506 K) and 350°F (450 K), respectively. Limit the use of metal linings to the upstream side of the burst disk.

3.4.5.1 TEMPERATURE EFFECTS ON PROPERTIES

The physical properties of the burst disk material over the range of operating temperatures shall be adequate to meet the burst pressure and service life requirements.

Aluminum and copper should not be used above 250°F (394 K), stainless steel above 600°F (589 K), nickel above 750°F (672 K). Inconel, however, can be used at 1000°F (811 K).

3.4.5.2 FATIGUE STRENGTH

The burst disk shall withstand the operating pressure and pressure oscillations of the application without failure by fatigue.

Prebulged disks may be operated at pressures as high as 90 percent of burst pressure, but this requires a static type of service. In general, restrict operating pressures to 70 percent of burst pressure.

The flat, coined-groove disk has been used much more in functional applications where long service is not required, and equivalent data are not available; however, the presence of a groove that would cause stress concentrations probably would reduce the above pressure values.

3.4.6 Installation

The burst disk installation shall not result in failure to meet the specified burst pressure.

Establish the need for the burst disk as early as possible in the system definition stage. Obtain recommendations from suppliers before the design is finalized.

In assemblies where the burst disk is welded in place, have the supplier of the burst disk do the welding. Avoid thick-to-thin weld sections closer than 0.13 in. (3.3 mm) to the groove of the coined-groove burst disk. In a formed-cup coined-groove burst disk, use the welded assembly shown in figure 31. For minimum burst tolerances, install the flat coined-groove disk so that the groove faces downstream and is on the same side as the weld. In burst-pressure sample testing of mechanically clamped disks, use clamping means for the test fixture identical to those used in the application.

The burst disk should not be exposed to any condition that might result in damage. Protect the burst disk from physical damage during assembly, test, storage, and use.

3.5 EXPLOSIVE VALVES

3.5.1 Body

The body configuration shall have low pressure drop and a shutoff capability requiring as small an actuator as possible.

Avoid discontinuities in flow passages. Intersect drill passages on centerlines with penetration to the point where full diameters blend. Use as few right-angle bends as possible. Provide a diaphragm thickness (fig. 33) adequate for the applied pressures but requiring as small a diameter actuator rod as possible. Use materials compatible with the fluids.

3.5.2 Actuator

The actuator shall generate an adequate force, with minimum loss of combustion gases.

Determine the amount of solid propellant necessary to shear the sealing membrane and actuate the valve to the open position. Make the housing, electrical connector, and seals strong enough for extremes of combustion energy. Limit storage temperature to less than +250°F (394 K).

3.5.3 Operational Problems

3.5.3.1 CONNECTOR LEAKAGE

The insulation material of the connector pins shall withstand explosive charge pressures without cracking or leaking.

Use epoxy resin for pin insulation because of its high strength and thermal properties. Avoid using glass or ceramic insulating material on detonating explosive charges. Use pin material with a thermal expansion coefficient as close as possible to that of the insulating material. For glass or ceramic insulating material, choose a high-nickel-iron alloy.

3.5.3.2 LEAKAGE OF ACTUATION GAS INTO SYSTEM

Actuation of the valve shall not introduce an excess amount of actuation gas into the valved system.

If actuation gas cannot be tolerated in the valved system, use the press-fit type of ram actuator. Use O-ring type of dynamic seals on the ram actuator only in applications where leakage of the actuation gas into the system can be tolerated.

3.5.3.3 SEALS

The valve seal shall be compatible with the fluid.

Use seals that will not deteriorate in the fluid, react with the fluid, take a permanent set, or cause corrosion. If necessary, establish compatibility by test under simulated use conditions. Use metal seals or welded joints when available elastomers are incompatible with the service fluid.

3.5.3.4 ESCAPE OF DEBRIS

Escaped debris shall not exceed allowable amounts.

Contain gross contaminants such as an end of a fitting or a diaphragm within the valve body. Use a filter in the downstream section of the valve body to remove debris that may be harmful to other components; use deflectors around filters to protect them from high-velocity particles. For systems that cannot tolerate debris, select valve configurations that do not generate debris (e.g., those shown in figures 33 and 36).

3.5.3.5 SHOCK

The explosive valve shall not transmit excessive shock to the vehicle.

Determine the magnitude of the shock created by operation of the explosive valve. Determine the effect of the shock forces on adjacent equipment in the valve application. Provide shock mounts as required.

3.5.3.6 ACCIDENTAL OPERATION

The electroexplosive device for initiating the valve operation shall be safe to store, handle, and use under the conditions of service.

The unit should not fire or be dudded by application of 1 ampere-1 watt for a specified period or by discharge of 25,000 volts from a 500-picofarad capacitor applied between bridgewire and case. Use the guidelines given in reference 73.

Use the specifications given in references 70, 71, 72, 74, and 75 to ensure a reliable explosive charge that may be assembled, inspected, handled, tested, and used safely.

3.5.3.7 TEMPERATURE EFFECTS

Valve response shall not be adversely effected by ambient-temperature extremes.

Determine the effects of ambient-temperature extremes on valve response and verify that system response requirements will be met under all conditions. If necessary, revise the design or select materials whose behavior is acceptable over the desired range.

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GLOSSARY

Term	Definition
actuator	device that converts control energy into mechanical motion to operate a valving element.
Belleville spring	truncated conical metal spring washer that can be designed to provide a negative mechanical spring rate.
bias spring	spring used to maintain a valve detail part in a preferred position while assembly is nonoperating.
bleed	continuous flow of gas through pilot (figs. 8, 9).
chatter	uncontrolled rapid seating and unseating of a valving unit, usually at low-flow conditions.
coined groove	narrow channel or depression stamped in a burst disk to provide localized thinning of material in a desired pattern.
Coulomb damping	dry friction damping; friction force is constant in magnitude but always directed opposite to the velocity.
cryogenic	fluids or conditions at low temperatures, usually at or below -150°C (123 K).
diaphragm	thin membrane that can be used as a seal to prevent fluid leakage and as an actuator to transform an applied pressure to linear force.
galling	progressive surface damage of mating surfaces resulting in increased friction and possible seizure.
hard anodize	production of a protective oxide film on a light metal (usually aluminum) by electrochemical means.
interflow	inlet and discharge of gas from the control-pressure region are controlled by a pressure-sensor-positioned 3-way valve that permits flow into and out of the control pressure region simultaneously. Relative values of P_S, P_C, P_R imply that gas flows from the larger parameter. (e.g., $P_S > P_C$, gas flows from P_S to P_C) (figs. 8 and 9).
lockup	all flow of gas stops when the regulated pressure exceeds the specified regulated value. In the pilot circuit, a sufficient increase in P_R can cause flow of gas through the pilot to cease (figs. 8 and 9).

Term	Definition
modulating	flow, pressure, or position is proportional to a sensed parameter and is infinitely variable within the regulated range.
negative gain	an increase in P_R causes an amplified decrease in P_C (figs. 8 and 9).
no bleed	under steady-state conditions, there is no flow of gas through pilot (figs. 8 and 9).
noninterflow	inlet and discharge of gas from the control-pressure region are controlled by a pressure-sensor-positioned 3-way valve that does not permit flow into and out of the control-pressure region simultaneously (figs. 8 and 9).
nonmodulating	the controlled variable (flow, pressure, etc.) cycles between limits.
pilot circuit	the flow-control elements (orifices, diaphragms, springs, etc) that, in combination with a pilot valve, control the operation of a larger valve.
pilot valve	a low-capacity valve that amplifies a low-power control signal to operate a larger valve.
positive gain	an increase in P_R causes an amplified increase in P_C (figs. 8 and 9).
reference load	a relative constant force that is used to modulate a valve within a desired value.
response	the capability of a valve to achieve 63 percent of a signaled position with minimum time delay.
restriction inlet	a flow path of reduced cross section or an orifice through which gas flows into the control-pressure region; gas is discharged through a throttling valve positioned by the pressure sensor (figs. 8 and 9).
restriction outlet	a flow path of reduced cross section or an orifice through which gas is discharged from the control-pressure region; gas flows into the control-pressure region through a throttling valve positioned by the pressure sensor (figs. 8 and 9).
shut off	a mechanism that stops the flow of fluid, or a condition of zero flow.
spring rate	change in spring force per unit change in length.
storable propellant	a propellant with a vapor pressure such that the propellant can be stored in a specified environment (earth or space) at moderate ullage pressures without significant loss over a specified period of time.
TIR	abbreviation for total indicated runout.

Term	Definition
ullage	amount that a container lacks of being full.
zero gain	P_C is held constant (figs. 8 and 9).

Symbol	Definition
A_A	actuator area
A_d	diaphragm effective area
A_i	inlet area
A_s	unbalanced seat area
B	bias-spring installed load
C	leakage constant = $\frac{2V}{\gamma GF}$
F	conversion factor for weight flow to volume flow
G	specific gas constant
g	acceleration due to gravity
L	reference spring load
P_A	atmospheric pressure
P_C	control pressure
P_{cr}	cracking pressure
P_R	regulated pressure
P_S	supply pressure
P_t	specified pressure
P_1	initial pressure
P_2	final pressure
T_1	initial temperature
T_2	final temperature
t	length of test

Symbol	Definition
V	volume of gas downstream of regulator
x	inlet valve stroke
Y	reference spring rate
Z	bias spring rate
γ	specific heat ratio

MATERIALS[1]

Designation	Identification
A-50, Aerozine-50	storable liquid propellant: a 50-50 mixture of unsymmetrical dimethylhydrazine (UDMH) and hydrazine per MIL-P-27402.
ceramic	an inorganic composition fired at high temperature to obtain a polycrystalline form.
Chempac	graphite-impregnated asbestos material manufactured by Johns Manville Corp.
CRES	corrosion resistant steel.
DC 55	silicone grease manufactured by Dow Corning Co.
elastomer	polymeric material that at room temperature can be stretched to twice its length and return quickly to its original length.
electroless nickel	nickel plate applied by nonelectrical means (chemical reduction process).
epoxy resin	class of thermosetting resins characterized by an epoxy group; widely used for adhesives, potting compounds, coatings, and laminated products.
FS 1281	flurosilicone grease manufactured by Dow Corning Co.
He	helium, pressurant per MIL-P-27407.
Inconel 718	International Nickel Co. designation for an austenitic nickel-base alloy.
Invar	International Nickel Co. designation for a nickel alloy with very low coefficient of thermal expansion.

[1] Additional information on metallic materials listed herein can be found in the 1972 SAE Handbook, Society of Automotive Engineers, Inc., Two Pennsylvania Plaza, New York, NY 10001.

Designation	Identification
Kel-F	trademark for a polymer of chlorotrifluoruoethylene manufactured by 3M Company.
LH_2	liquid hydrogen, propellant grade per MIL-P-27201A.
LO_2	liquid oxygen, propellant grade per MIL-P-25508D.
Monel	International Nickel Co. designation for a nickel-copper alloy.
Mylar	trademark for polyethylene terephthalate film manufactured by the E. I. duPont Co.
Nylon	thermoplastic polyamide.
Ni-Span-C	tradename of International Nickel Co. for an iron-nickel-chromium alloy that can be heat treated to produce an essentially constant modulus over a wide temperature range.
RP-1	high-energy kerosene, propellant grade per MIL-P-25576.
rubber	an elastomer, either a synthetic or a natural compound obtained from the hevea brasiliensis tree.
TEAB	a pyrophoric mixture of triethylaluminum and triethylborane.
Teflon	trademark for polymerized tetrafluoroethylene manufactured by the E. I. duPont Co.
17-7 PH	semi-austenitic precipitation-hardening stainless steel per MIL-S-25043.
300 series	austenitic stainless steels per QQ-S-763.
440 C	martensitic stainless steel per QQ-S-763.
110-0 3003-0 5052-0 6061-0	wrought aluminum alloys of soft temper.

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ENVIRONMENT

SP-8005	Solar Electromagnetic Radiation, Revised May 1971
SP-8010	Models of Mars Atmosphere (1967), May 1968
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SP-8013	Meteoroid Environment Model–1969 (Near Earth to Lunar Surface), March 1969
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SP-8001	Buffeting During Atmospheric Ascent, Revised November 1970

SP-8002	Flight-Loads Measurements During Launch and Exit, December 1964
SP-8003	Flutter, Buzz, and Divergence, July 1964
SP-8004	Panel Flutter, Revised June 1972
SP-8006	Local Steady Aerodynamic Loads During Launch and Exit, May 1965
SP-8007	Buckling of Thin-Walled Circular Cylinders, Revised August 1968
SP-8008	Prelaunch Ground Wind Loads, November 1965
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SP-8012	Natural Vibration Modal Analysis, September 1968
SP-8014	Entry Thermal Protection, August 1968
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SP-8030	Transient Loads From Thrust Excitation, February 1969
SP-8031	Slosh Suppression, May 1969
SP-8032	Buckling of Thin-Walled Doubly Curved Shells, August 1969
SP-8035	Wind Loads During Ascent, June 1970
SP-8040	Fracture Control of Metallic Pressure Vessels, May 1970
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SP-8053	Nuclear and Space Radiation Effects on Materials, June 1970

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SP-8063	Lubrication, Friction, and Wear, June 1971
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SP-8072	Acoustic Loads Generated by the Propulsion System, June 1971
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SP-8018	Spacecraft Magnetic Torques, March 1969

SP-8024	Spacecraft Gravitational Torques, May 1969
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SP-8086	Space Vehicle Displays Design Criteria, March 1972
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SP-8098	Effects of Structural Flexibility on Entry Vehicle Control Systems, June 1972
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